NASA CR-66174

DEVELOPM
PROTOTYPE MASS MEA NT SYSTEM
FO

PREPARED UNDER CONTRACT NO. NAS 1-5999

by

Biotechnology

Lockheed Missiles & Space Company

Sunnyvale, California

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

LANGLEY RESEARCH CENTER

LANGLEY STATION

HAMPTON, VIRGINIA

NASA CR-66174
28 October 1966

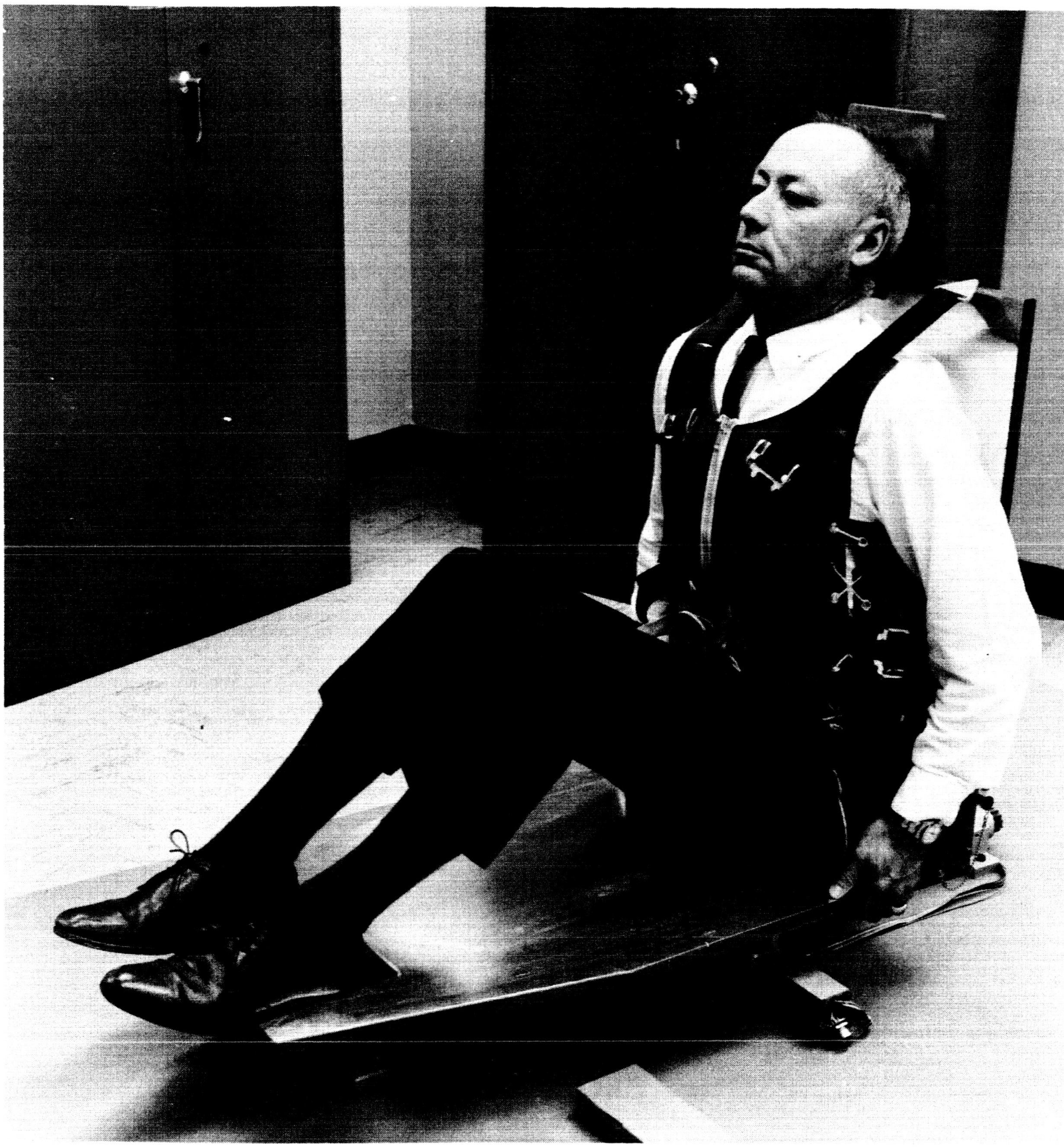
DEVELOPMENT OF PROTOTYPE MASS MEASUREMENT SYSTEM FOR SPACEFLIGHT

PREPARED UNDER CONTRACT NO. NAS 1-5999

by
Biotechnology
Lockheed Missiles & Space Company
Sunnyvale, California

R. B. MAINE
A. L. WEITZMANN

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
LANGLEY RESEARCH CENTER
LANGLEY STATION
HAMPTON, VIRGINIA



Prototype Mass Measurement System

LIST OF CONTRIBUTORS

Name	Area of Contribution
R. B. Maine	Project Direction
A. L. Weitzmann	Design
A. Shafran	Design
R. W. Joy	Design
M. S. Hubschmitt	Design
H. T. Fisher	Human Engineering
W. H. Monson	Test
R. S. Thomas	Biophysics
H. Giarretto	Biophysics
F. E. Riley	Integration
Consultant	
Brandt Clark, Ph. D.	Vestibular Functions

NASA TECHNICAL MONITOR

Bruce Conway

PRECEDING PAGE BLANK NOT FILMED.

CONTENTS

CONTENTS (Cont'd)

ILLUSTRATIONS

ILLUSTRATIONS (Cont'd)

TABLES

SUMMARY

As man extends his ability to stay in space for long duration missions, one very important physiological data point on his well-being will be an accurate method of determining body weight in the zero-gravity environment. The need for a Mass Measurement System requires the development of a suitable device to effect mass determination not only of flight crew personnel, but other objects in the spacecraft whose mass must be determined for various research experiments. The purpose of this program is to study, design, fabricate, and test a Mass Measurement System (MMS) prototype suitable for both zero- and partial-gravity environments.

Of the many systems studied, the oscillating spring-mass system was selected for hardware development; its operating principle is based on the simple harmonic motion of an oscillating mass coupled with the precise timing of the oscillations.

The development and testing of the MMS hardware proved that this principle was sound and that accuracies of less than ±0.25 pounds were possible in the weight range of 5 to 40 pounds and ±0.50 pound accuracy could be achieved in the 41 to 250 pound range. The accuracy of human subjects was within ±1.0 pound.

System studies indicated that use of the MMS hardware to measure other physiological parameters was desirable to more fully utilize the equipment. Investigations show that with minor changes to the basic MMS, the device can be used to determine human threshold accelerations in the range of 5 to 10 cm/sec^2.

The complete MMS will pass through a 32 in. diameter port, with the back in the folded position, and is 49 inches in length. The design goal for a system weight of 25 pounds was not accomplished. However, a MMS for a spacecraft or space station could be fabricated within the 25 pound requirement. The total volume requirement of 3 cubic feet was met under certain conditions of storage configuration. Studies indicate that a true spacecraft version of this system will be within the 3 cubic feet requirement.

Two carriage systems, using different track and bearing concepts, were designed and fabricated. Tests show that both systems are within the design goals of mass determination accuracy.

To provide a simple means of converting the oscillation time readout in seconds directly to pounds, an existing LMSC computer program was used to develop conversion tables from the calibration test runs of both animate and inanimate objects as well as for human subjects.

With the MMS concept proven within the accuracies required, the next step in the hardware development would be the fabrication of a KC-135 flight prototype verifying performance characteristics while flying a zero – g parabola, and the preliminary design of a spacecraft/space station oriented mass measurement system.

INTRODUCTION

The NASA-Langley Research Center, recognizing future manned space program requirements, directed the Lockheed Missiles and Space Company (LMSC) to develop prototype equipment specifically designed to monitor mass changes of astronauts, and to determine the mass of spacecraft materials during orbital flight. Adaptability of the Mass Measurement System (MMS) to other Apollo Application Program experimental objectives, such as measurement of human threshold acceleration, was investigated.

The specific purpose of the current program was to develop a prototype MMS capable of weighing objects in the 5 to 40 lb range (±0.25 lb), and in the 41 to 250 lb range (±0.50 lb).

A logical sequence of development was followed during the course of this project. Conceptual designs and laboratory experimental work were reviewed and a comparison analysis conducted for the optimum technique selection. Configuration studies followed that analyzed design concepts for major subsystems, particularly those concerned with the pallet, carriage, restraint, and electronic timing method.

Conclusion of the preliminary investigations permitted initiation of the detail engineering design phase and fabrication of the selected system. Sufficient engineering data developed early in the project permitted development of a test plan and incorporation of test requirements into the final design. Preliminary performance specifications furnished the basis for the performance testing of the MMS. Results of the test program are documented in this report.

A major objective was to develop the prototype system in such a way that the design could be utilized in the development of a flight prototype to be flown on a KC-135 zero-g flight parabola. Preliminary design analysis for the overall design, as well as the structural design, was performed for flight conditions. Recommendations of structural modification to the present prototype MMS are made.

This report treats the individual development steps and provides test results and recommendations as to the future course of action. Data provided will permit the early initiation of a flight-type MMS and the completion of that phase sufficiently in advance to allow time for system integration into the NASA Manned Space Program.

CONCEPT REVIEW AND ANALYSIS

This section presents results of an analytical study of different theoretical methods for measuring mass in a weightless environment. Candidate systems are established and a comparative analysis performed.

Problem Statement

Manned space programs now being projected for the future contemplate missions of increasing duration in which (1) the body mass of the astronaut will be subjected to significant and probably vital changes, (2) the sensitivity of the otolith mechanism may increase as a result of the absence of linear acceleration and gravity, (3) the apparent change in position of an object in the visual field (the oculogravic illusion) may be affected.

Experiments have been proposed by NASA and other principal investigators that will investigate these hypotheses. This project was initiated to develop a versatile prototype MMS designed to meet these experimental objectives.

Mass measurement. – During prolonged missions the body mass of the astronaut may be subject to significant and probably vital changes (either gain or loss). Loss of mass may result from deconditioning and atrophy of muscle tissue, and body fluid loss by diuresis. In addition, there is interest in monitoring mass changes of spacecraft materials, both internal and external, animal experiment mass changes, mass of recovered satellites for post Apollo missions, mass of spacecraft expendables (e.g., food) and waste products (CO_2 canisters), and mass changes of fully-equipped astronauts during extravehicular activity. It is desired to monitor changes for astronauts within ±1.0 lb. For smaller masses, 5 to 40 lb, ±0.25 lb should be adequate. For masses 40 to 250 lb, ±0.50 lb is adequate.

Linear acceleration. – Man's ability to react and position himself in relation to the gravitational forces of his environment depends to some degree upon the sensory properties of the otolith mechanisms of the inner ear. When signals to the brain from this organ conflict with those from the eyes or the other somatic sensory modalities, confusion may occur accompanied by vertigo, motion illusions and possible motion sickness. Consequently, changes in the degree of the sensitivity of the otolith organ to acceleration may be an important determinant of man's capability for task performance and for accomplishing critical spacecraft guidance maneuvers.

To quantitatively evaluate the sensitivity of the otolith mechanism it is necessary to expose the subject to low accelerations in the range of 5 to 10 cm/sec^2 (0.165 to 0.32 ft/sec^2).

Rotary acceleration. – There is interest in measuring the functional state of the semicircular canals of the statokinetic labyrinth during orbital flight in order to determine possible changes in their sensitivity as the result of prolonged weightless exposure. The oculogravic illusion, e.g., the apparent change in position of an object in the visual field is of particular importance. The significance of the oculogyral-coriolis illusion is investigated by producing head movement during rotation. Thresholds and maximum tolerance can be measured objectively and subjectively. Stimulus threshold determinations are achieved by producing appropriate angular accelerations for each astronaut within the range from 0.1 deg/sec^2 to 10 deg/sec^2. For suprathreshold stimulation of the semicircular canal, data may be obtained for selected accelerations and decelerations from 0.6 deg to 10 deg/sec^2, or they may be obtained from rapid accelerations to 3, 5, and 10 rpm in 3 to 6 sec. Following a one-minute interval, the rotational device must be decelerated to zero velocity in 6 sec, and reading of the apparent rotation again taken.

Candidate Mass Measurement Systems

LMSC has conducted an analytical study of seven different theoretical methods for measuring mass in a weightless environment. The analytical study was followed by the development of three laboratory models, two based on the principle of an oscillating spring-mass and one based on the constant linear acceleration principle. Extensive calibration runs and testing under various loading conditions have led to the recommendation that the prototype MMS to be developed for NASA-Langley be based on the principle of spring-actuated oscillating mass.

The overall design considerations were that the MMS must be capable of weighing objects in the 5 to 40 lb range (±0.25 lb) and in the 41 to 250 lb range (±0.50 lb). Adaptation of the MMS to other experimental measurements would be highly desirable. The system must be capable of operating during a zero-g trajectory in a KC-135 aircraft and be capable of partial disassembly or folding for stowing in a minimum space.

In addition to these general design criteria, specific areas to be investigated were:

- Effect of the restraining harness on the mass determination
- Capability of the MMS to weigh miscellaneous spacecraft parts including partially filled containers of liquid
- Effect of eccentric pallet loading and lateral and longitudinal tilt of the unit
- Design verification tests
- Effect of varying the number of oscillations to be timed.
- Accuracy of system to meet design goal of ±0.25 lb on a rigid mass of 20 lbs
- Accuracy of system to meet design goal of ±0.50 lb on a rigid mass of 100 lbs
- Evaluation of data readout techniques

A review of different techniques for determining the mass of an astronaut during spaceflight was conducted. Obviously, the most common means of weighing on earth—the beam balance and the calibrated spring scale cannot be used in an environment where there is no gravitational attraction, at least not without modification. Other properties of matter, therefore, will have to be used. It is possible to determine the body mass using X-rays since the body's opacity to X-rays varies with the mass of the body. However, this will require complex, sensitive, and delicate apparatus, and an extensive development program. Moreover, even "soft" X-rays may have undesirable cumulative effects on the human body. It will be less hazardous to determine mass by inertia — another of its properties. Some of the conceivable means of using inertia to determine mass are as follows:

Torsional pendulum. — This device has a frequency

$$f = \frac{1}{2\pi}\sqrt{\frac{\tau}{I}}$$

where

$$\tau = \frac{\text{restoring torque}}{\text{angle}}$$

$I = Mr^2$ = Product of mass times the square of the radius of the path of the mass center.

Since the frequency is inversely proportional to the radius, small changes in location of the mass center will cause appreciable errors unless the radius is very large. However, an apparatus with a very large radius will exceed the size limits of a spacecraft.

Centrifuge. — The equation for centrifugal force is:

$$F = M\omega^2 R$$

where

M = Mass

ω = angular velocity

R = radius of the path of the mass center

Since (R) enters this equation linearly, a small error in assumed mass-center location will not be as serious as in the case of the torsional pendulum. However, angular velocity enters as a squared term and, therefore, will have to be closely controlled. It will not be difficult to measure centrifugal force, but a seat or couch having radial freedom of movement along one arm of the centrifuge will be required.

Impulse-Momentum. — The equation for this impact action is:

$$F\ (\Delta T) = M\ (\Delta V).$$

where

F = Force which is applied over a time period ΔT.

M = Mass which is being accelerated from its initial velocity to its final velocity ΔV.

In other words, $F\ (\Delta T)$ represents an impulse which changes momentum $M\ (\Delta V)$. Both F and (ΔT) will be difficult to determine because of their transient nature. The presence of random spacecraft acceleration will adversely affect accuracy; therefore, precautions will be required to ensure that no such disruptive forces exist during astronaut mass determination. Further, in a zero-g environment, an astronaut will not be bedded down firmly on the apparatus at the first instant when acceleration starts. Even though he is strapped or clamped into the apparatus, his weightless configuration relative to the apparatus will not be identical to that which his body will assume after inertia forces are present. These considerations make the impulse-momentum method quite impractical.

Conservation of momentum. — The equation for this phenomenon is:

$$MV_1 + mv_1 = MV_2 + mv_2$$

It represents a controlled collision in which a smaller known mass (m) traveling with a known initial velocity (v_1) strikes the unknown mass (M) which is stationary $(V_1 = 0)$. A device must be provided to lock the two masses together upon impact so that $V_2 = v_2$. The equation thus becomes:

$$M = (m/v_2)\ (v_1 - v_2).$$

To determine the unknown mass (M), the final velocity (v_2) must be measured. This can be accomplished by timing the passage of the coupled masses over a known distance. The disadvantages of this conservation of momentum method are that an appreciable auxiliary mass (m) is needed, and that a considerable acceleration will be caused by the impact. Since the presence of spacecraft acceleration can cause inaccuracies, mass determination should not be attempted unless such forces are absent.

Linear acceleration. — This method is represented by the equation:

$$F = Ma$$

where

F = Known, constant force

M = Mass

a = acceleration

This acceleration can either be measured directly, using an accelerometer, or indirectly by determining the time (t) to traverse a known distance (s) as:

$$a = \frac{2s}{t^2}$$

The time must be measured very accurately since it is a squared term. If any random spacecraft accelerations are present, they can cause inaccurate results.

The bedding-down aspect previously noted (the impulse-momentum system) does not apply very strongly to this system because it involves a far longer period of acceleration, and the bedding-down will take place at the start of motion. Further, the acceleration will be low, and consequently, the bedding-down effect will be smaller in magnitude. LMSC initiated a verification of this technique by constructing a laboratory device. This unit consisted of a plywood cart with ball-bearing wheels. The wheels ran on a track which is horizontally level. To achieve the desired constant force, a Negator spring was used. Measurement of time was accomplished by using an electronic counter to measure elapsed time as the cart traveled over a known distance. The start and stop signals were originated by a cam operated switch.

These tests proved the feasibility of using a linear acceleration device with an electronic timer to determine a man's mass in accordance with Newton's second law of motion. It was determined that the subject must take care to remain rigid during the test run to avoid disturbing his inertia.

Inertia beam balance. — A parallel platform type of beam balance (i.e., a commercial weighing scale) can be employed by applying a momentary acceleration in line with the knife edge so that the resulting inertia force acts to replace gravitational attraction in the weighing function. The momentary acceleration can be obtained by moving the balance upward, using a man-powered lever. This system has the great advantage that no additional instruments of any kind are required since there is no need to measure time, force, acceleration, etc. The presence of a random g force will tend to reduce accuracy slightly, but its existence can be detected by making a trial balance without any subject on the platform. In order to overcome the undesirable effects of the bedding-down problem, a rather long acceleration stroke will be required, perhaps with a provision for performing the balance only during the latter part of the stroke.

Oscillating spring-mass system. — The frequency of an oscillating spring-mass system using two springs is:

$$f = \frac{1}{2\pi}\sqrt{2K/M}$$

where

f = frequency (cps)

K = known spring constant

M = unknown mass

To determine mass, one need only provide a single-degree-of-freedom (i.e., straight back-and-forth linear motion) mounting for the unknown mass, provide a spring system to keep the mass oscillating in simple harmonic motion and measure the system frequency. Timing can be performed with great accuracy because an average of a number of cycles can be obtained. Further, the presence of random g forces will have little or no effect on accuracy. With respect to the bedding-down effect, the spring-mass system will not be quite as good as the straight linear acceleration system. However, the length of this apparatus can be minimized.

The oscillating spring-mass system was evaluated by using a laboratory model similar to the one used for the linear acceleration runs. The major change was replacing the Negator device with fore and aft helical tension springs. Runs were made on a track and on a suspended platform. Use of the cart with ball-bearing wheels running on the straight track eliminated the pendulum term from the equation relating time and mass. It also eliminated any sidewise motions, allowing only one degree of freedom.

Comparison Analysis

Of the candidate mass measurement systems that have been reviewed, four appear to be feasible. A comparison of these systems with respect to some of the critical considerations is shown in Table 1. Using the tabulated comments as a basis of judgment, the following conclusions can be drawn:

Centrifuge. — This system provides an excellent way of determining mass if a facility the size and weight of a centrifuge is justified for other pruposes; however, some crew members may be excluded from the centrifuge for control purposes, so other means of weighing them will be required.

Inertia beam balance. — This method is the most straightforward since secondary quantities (e.g., time, acceleration, velocity, or force) need not be measured. Further, acceleration need not be closely controlled. This method is preferable except that simulation of its response in a zero-g environment is difficult and elaborate simulation methods will be required.

Table 1

CRITERIA OF COMPARISON FOR ALTERNATE MASS MEASUREMENT SYSTEMS

Mass Measurement System	Bedding Down	Apparatus Simplicity and Alternatives	Suitability for Ground Evaluation	Effect of Difference in Bearing Friction Between Ground and Orbit	Operational Considerations
Linear Acceleration (F = Ma)	Little effect	Electronic timer moderately complex; proposed on-board computer could perform function	Ground evaluation satisfactory; may require recalibration for zero-g condition	Could be appreciable; ground calibration chart requires alteration for orbital use	One shot; a 1-man operation
Oscillating Spring-Mass System	Greater adverse effect on accuracy than linear acceleration	Electronic timer moderately complex; proposed on-board computer could perform function; if electronic timer fails, a stopwatch usable (less accuracy)	Ground evaluation satisfactory; may require recalibration for zero-g condition	Could be appreciable; ground calibration chart requires alteration for orbital use	One shot; a 1-man operation
Inertia-Beam Balance Technique	Phased-stroke solution to potential problem	Simplest system	Ground simulation difficult; elaborate means required	None	Trial and error; possible in 2-man operation
Centrifuge	No problem	Relatively simple (if centrifuge exists for other purposes)	Ground calibration setup can simulate zero-g condition	None	Impose scheduling constraints, unless performed concurrently with other centrifuge function; involves long operating cycle

Linear acceleration. – The electronic timing apparatus (or accelerometer) required for this method is complex, but an on-board computer can be used to perform these functions. The length of track required and the mechanical complexity associated with assuring that the initial velocity is zero makes this system cumbersome. Accuracy might be increased by repeating runs and averaging.

Oscillating spring-mass system. – Almost the same comments as those for the straight-linear-acceleration method apply to this system. However, this system has an advantage because, in the event of failure of electronic timing apparatus, a mass determination can be accomplished by using a stop watch even though it will be less accurate. Further, it is virtually unaffected by small gravity or centrifugal forces at random orientations. This system, therefore, is recommended at this time for mass measurement purposes.

Based on the review of various techniques and laboratory verification of three models, the recommended oscillating spring-mass measurement system general requirements are:

- Simple harmonic motion, with a carriage and track limiting it to one degree of freedom should be employed.
- A rigid pallet mounted on low friction, high capacity bearings should be provided to eliminate variable friction and damping.
- A restraint system must be provided to prevent movement of the subject relative to the carriage.
- The MMS must be designed to use common parts wherever feasible for the mass determination, and the acceleration threshold determination.
- Weight, size, and power should be held to a minimum.

CONFIGURATION STUDIES

With the basic concepts and requirements established for the MMS, the first task of the program was to study the various hardware design approaches which would best satisfy the known requirements. All aspects of the candidate designs received careful consideration with emphasis being placed on strength-to-weight ratio, selection of materials which will not outgas toxic contaminants, minimum volume, low electrical power requirements, materials suitable for vacuum operation, simplicity of operation, future adaptability, and high reliability.

The device has been divided into two basic design areas: (1) the pallet or device for holding the mass to be determined and, (2) the carriage system which forms the hardware that allows the complete MMS to oscillate.

Pallet

A review of the MMS requirements indicated that two basic pallet designs would provide a base from which man and other masses, both animate and inaminate, could be restrained. One design employs the use of small diameter tubes welded together to form a truss structure. The second design is based on the use of honeycomb sandwich materials.

Truss structure. – Strength-to-weight ratio studies indicate that the use of small diameter aluminum tubing arranged to form a truss structure will provide a very stable platform. An aluminum skin welded to the truss structure forms seat and seat back. These skins are stiffened by diagonal beads to prevent canning and are perforated with lightening holes to reduce overall weight. This configuration is shown in fig. 1.

The seat back folds forward from a pivot fitting located at the aft corners of the truss. Quick-release type lock pins are used to position the back in either the upright or folded position. This feature does two things. First, it allows the device in the folded position to pass through a 32 in. diameter port, a program requirement. Second, when folded, the aft surface of the back provides a pallet which, when fitted with a restraint device, will lock smaller objects in place for mass determination.

Lugs are provided on the truss structure to secure the restraint system for human subjects. These fittings are of the quick-release type to ensure ease of restraint system removal.

The ring located on the lower surface of the pallet is the interface point between the pallet and the carriage assembly. A plate welded to the ring will provide a universal bolting surface to which varied carriage designs can be fastened.

The calculated weight of the truss structure is 9.70 lb and has a volume of 2.75 cubic feet.

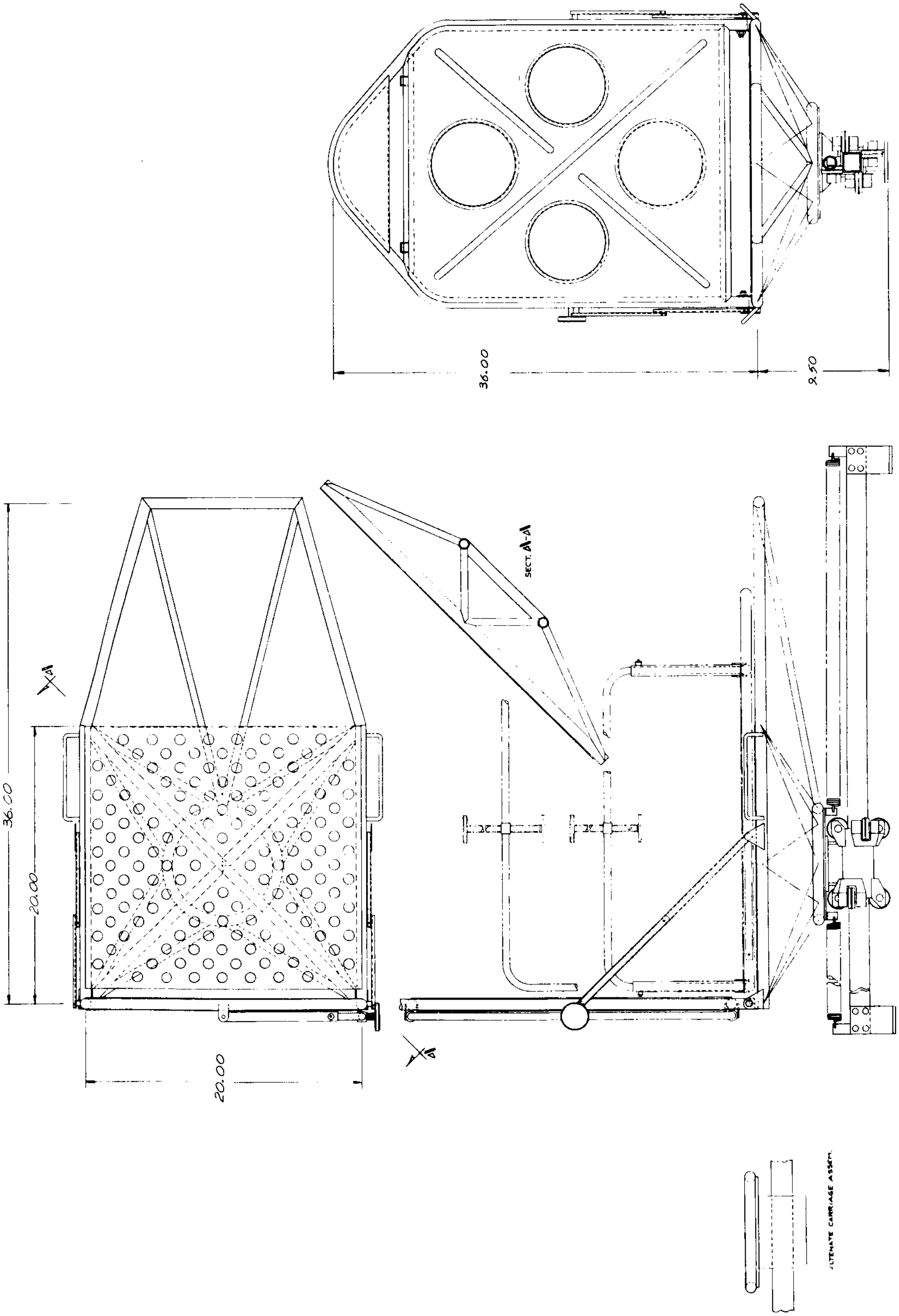


Fig. 1 Pallet Employing Aluminum Truss Structure

Honeycomb structure. — The second design studied employs the use of honeycomb sandwich panels. Preliminary stress analysis indicated that the inherent mechanical properties of aluminum honeycomb are particularly well-suited to the structural requirements of the pallet.

The edges of the honeycomb panels, which form the seat and back of the pallet, are reinforced with aluminum channels bonded in place. The carriage attachment ring is located in the center of the seat approximately at the center of gravity of a seated subject. The ring is bonded in place, with the honeycomb cells under the flanges of the ring. A bolt circle fitted with nutplates on the bottom flange of the ring provide attachment points for the carriage assemblies.

Attach points for the restraint system and clamping device, for the smaller masses, are reinforced by the bonding of aluminum channels and bars between the two skin panels during the fabrication of the honeycomb sandwich. A back attachment plate is mounted to each side of the aft edge of the seat. A pivot point in the plate will allow the back to fold forward and lock in place. The aft surface of the back, in the folded position, provides a surface to which the smaller mass clamping device is mounted. The configuration is shown in fig. 2.

Carriage

In order to achieve the desired accuracy, a one-degree-of-freedom carriage is essential since lateral pitching, or yawing motions would cause inconsistent and variable results. In effect, this requires a bearing system having small clearances. Among the candidates considered were sliding bearings employing materials such as Oilite or Teflon air bearings, magnetic support systems and ball bearings. Sliding bearings would probably have excessive friction; in the case of Teflon the problem of cold flow would also be encountered. The ultimate use of this mass measurement system for extra-vehicular activities makes use of air bearings too complex. Magnetic support systems are either heavy, or complex, or both. Ball bearings were selected for this application because of their low friction, high load carrying capacity, and moderate weight.

Another major decision was whether to have the bearings ride on one shaft or two. In order to permit only one degree of freedom, the carriage design had to provide for resistance to moments tending to rotate the carriage around its axis of travel. This could be accomplished by using two parallel shafts, with bearings riding on each of them. However, considering the strength and rigidity required of the shaft(s) when acting as a beam in a one g environment, for a given rigidity, one shaft would be lighter than two. Accordingly, this configuration was chosen.

Since only one shaft was to be used, this meant that it had to provide a means for preventing the bearing from rotating on it. One of the commercially available units meeting this requirement is the Saginaw, which employs a linear outer ball race riding on a splined inner race, or shaft. Another unit is the Turnomat, which uses a linear outer ball race riding on a square inner race, or shaft. Although the latter unit provided a ball clearance adjustment which the former unit did not, the Saginaw unit was selected because of its lighter weight. Both of these units are susceptible to intrusion of dust and dirt into the ball bearings, since the shaft itself is the inner race. Therefore, it was deemed essential to provide the Saginaw shaft with a protective bellows.

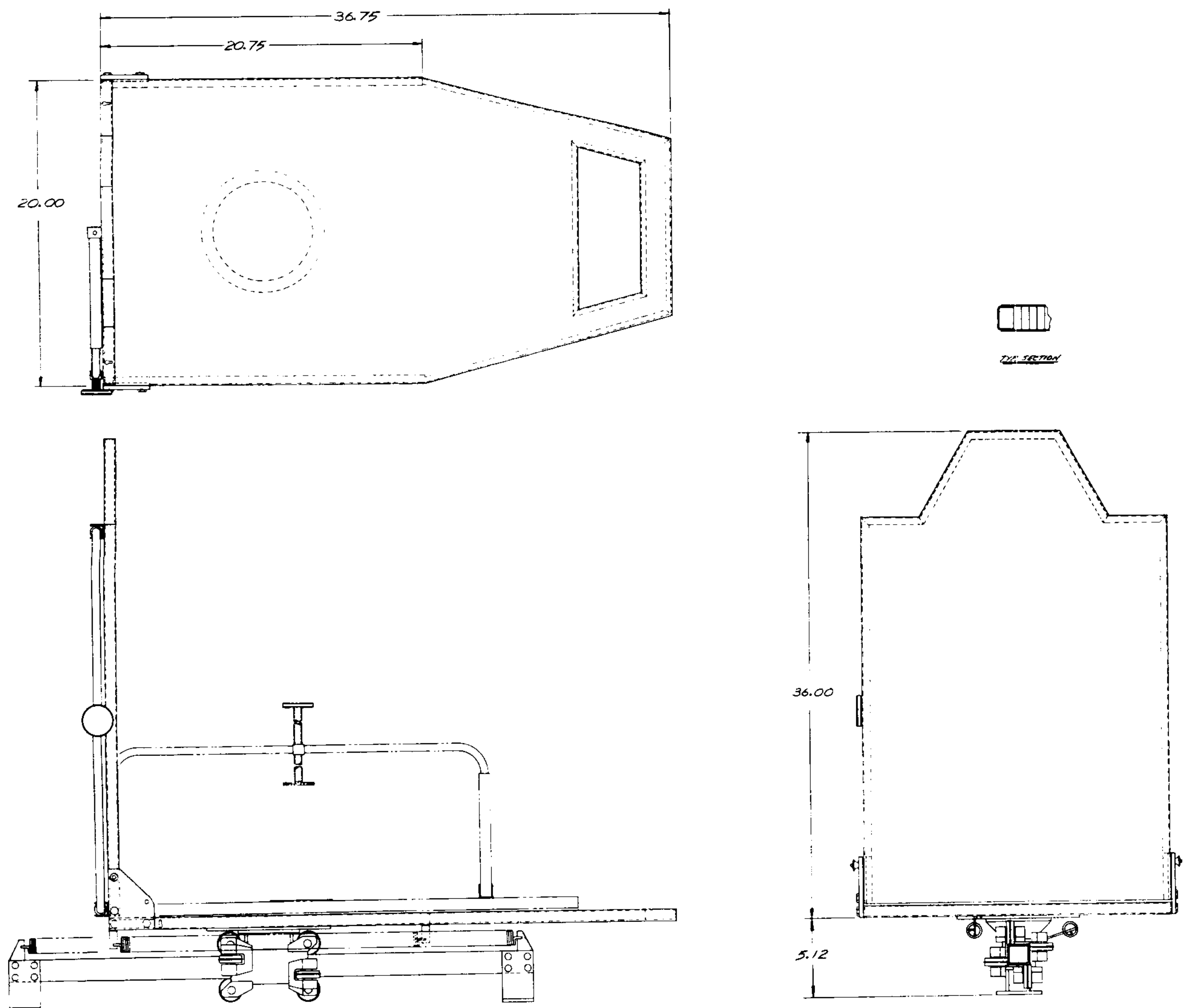


Fig. 2 Pallet Employing Honeycomb Sandwich Panels

Another equally good candidate configuration was conventional type ball bearings. By using a square shaft, a carriage could be designed to permit conventional ball bearings to ride on each of the four sides of the square shaft, thereby enabling the carriage to sustain twisting moments.

Choice of springs. — The previously built LMSC laboratory prototype mass measurement device employed springs having a constant of approximately 2.7 lbs per inch, and these springs were satisfactory not only for weighing people, but small objects as well. Therefore, it was decided to use approximately the same spring constant. Stainless steel was selected as the material. In order to determine other spring parameters, calculations were made and a tabulation was prepared (see Table 2). In order to prevent twisting moment from being applied to the carriage it was decided to locate the springs directly on the centerline, although this would necessitate increasing the length and the height of the device somewhat. The spring design which was chosen was similar to the one in the center of the tabulation.

Electronic Timing Device

To measure the time for the pallet to travel a discrete number of cycles requires a sensing device to indicate pallet position, a cycle counting device, and a time counting system.

The sensing device has to generate a signal when the pallet passes a particular point in its cyclic travel. This signal is used to start the time counting system and pulse the cycle counting device. The cycle counter must be able to accept the sensor pulses and count a discrete number after which it must generate a pulse to stop the counter. Thus, the time interval of a discrete number of pallet cycles can be read-out on the time counter.

To detect a weight difference of 0.25 lb in the desired range, the timing device should provide a total readout to within .001 second. The accuracy of such a system should be 10 times greater than the desired readout accuracy, or ± .0001 second.

The design of a system for measuring pallet travel time was considered from the standpoint of accuracy, simplicity, size, weight, and cost. A breakdown of the systems evaluated follows.

Sensing devices. — Sensing devices examined included: cam actuated switch, cadmium sulphide photoconductive cell, and photo transistor. A cam actuated switch met nearly all of the desired characteristics for a sensing device. A major objection to its use was switch-bounce inherent in most switch designs. A cadmium sulphide photo conductive cell was rejected because of its relatively long recovery time (on the order of 1 second). To use this device, its signal would have to be amplified and chopped in order to use the low-end of its variation. Of the three devices considered, the photo transistor exhibited the most favorable data. It can be procured in a small lightweight container; requires little additional circuitry to use; has an acceptable rise and fall time (on the order of microseconds); and is relatively inexpensive.

Table 2

SPRING COMPARISON CHART

No. of Turns	Wire Dia. (in.)	Length of Solid Coil (in.)	Static Length of Each Spring (in.)	Mean Coil Dia. (in.)	Static Length Two Springs End to End (in.)
128	.109	14	24	.859	48
64	.125	8	18	1.25	36
32	.148	4.74	14.74	2.0	29.48
16	.169	2.7	12.7	3.0	25.4
8	.193	1.54	11.54	4.5	23.08

Note: Assumed parameters: Spring constant = K = 2.7 lb/in.
Initial force = 50 lb, Max. stress = 80,000 psi

Cycle counters. — Cycle counters examined included: rotary solenoid step switch, mechanical counter, and electronic ripple counter. The rotary solenoid stepping switch is simple in operation and readily available, but its main disadvantages are that it is very noisy and has a delay time on the order of milliseconds. The repeatability of such a device also appears to be in the millisecond range. A mechanical counter could be arranged to trip the electronic timing counter, after a discrete number of pallet cycles, but the complexity of such a system and the apparent difficulty in achieving the desired accuracy, makes it unattractive for this application. The most desirable method for counting pallet cycles would be an electronic ripple counter. With four bistable multivibrators, properly gated, from 1 to 16 pallet cycle impulses could be counted. They could be triggered by the photo-transistor output and their output made to trigger an electronic counter. As integrated circuit components, the ripple counter could be made small and light, with an accuracy in the order of microseconds, or better.

Timing device. — An electronic timing device appears to be a good choice mainly because of the accuracy obtainable and its convenient readout characteristics. Hewlett-Packard's Model 5322L counter can provide an accuracy of ± .0001 seconds and its use in this system would be acceptable.

System selection. — The system selected utilized a photo-transistor, a bistable multivibrator ripple counter gated to count a discrete number of pallet cycles, a Hewlett-Packard Model 5322L counter, and a Lectrograph power supply.

Anthropometric Considerations

General design requirements. — The mass measurement device has been designed to accommodate the typical flight crew population of pilot and scientist astronauts whose bodily dimensions fall within the middle 90 per cent of the Air Force flying population (refs. 1 and 2). Design values for all body geometric dimensions, therefore, fall between the 5th and 95th percentiles of the full range of user personnel. Although the astronaut population size, as a general rule, is somewhat less than the total middle 90 percent of the flying population, certain astronaut trainees have exhibited body dimensions equal to, or greater than, the range limits. In addition, personnel involved in the ground and air flight test program may be larger than the typical astronaut population. Thus, the prototype mass measurement system has been designed to accommodate the 5th to 95th percentile range of user personnel. It may be possible to reduce design dimensions of the flight MMS slightly to accommodate a very specific range of astronaut/user personnel in order to reduce weight and volume.

Of particular interest in the shirt sleeve clothing complement to the design of the MMS are the constant wear garment (CWG) sandals. These lightweight sandals are worn with the CWG for shirtsleeve operations in the Apollo CM and LEM to restrain the astronaut and assist him in performing various subsystem and experimental tasks. Therefore, the utilization of the CWG sandals had to be considered in the design of the pallet foot rests. Both width and length of the foot rest were evaluated to determine minimum design dimensions sufficient to accommodate a 95th percentile man wearing the CWG sandals.

Basic body geometric dimensions necessary to design the mass measurement device pallet are presented in Table 3. No consideration was given to design of the pallet for a pressure suited subject since all measurement activities are to be confined to the interior of the CM or LEM during normal operations.

The mass measurement device and associated controls, harness fittings, and positioning references have been designed to accommodate the range of astronaut body movements necessary to actuate, manipulate or control the various man-machine interface hardware. This concept provides for minimum participation of the experimenter during experiment set-up, thus reducing the requirement for two personnel (experimenter and subject) simultaneously during the initial set-up phase.

Center of Gravity. – The center of gravity of the human body is a fundamental parameter which enters into all computations involving body movement and rotation. These computations arise in the analysis of total body motion under conditions of weightlessness, where body movement of an astronaut is easily produced by his own or an internal force as in the case of the mass measurement device.

In order to determine the point of the center of gravity for an individual, a body coordinate system was utilized for basic reference. The orthogonal axis system is defined by the intersection of the three principal planes of the body passing through the center of gravity of the body as shown in fig. 3. The Z-axis is formed by the intersection of the sagittal plane and the frontal plane. The Y-axis is formed by the intersection of the frontal and transverse planes and the X-axis is formed by the intersection of the sagittal and transverse planes. The orthogonal axis system is referenced to the body as shown in fig. 4. The location of the center of gravity of the body is measured along the Z-axis from the top of the head, L(Z); along the X-axis from the back plane, L(X); and along the Y-axis from the anterior superior spine of the ilium, L(Y). However, since body symmetry with respect to the sagittal plane was assumed. L(Y) is defined as equal to one-half the bispinous breadth.

The body of the astronaut positioned on the mass measurement pallet is illustrated in fig. 5. This position is referred to as sitting, legs extended with thighs slightly elevated. The literature was searched to determine if a center of gravity study had been conducted with the individual positioned as in fig. 5; however, no exact duplication of this body position could be found. Center of gravity body positions have been studied, however, which were fairly similar to the pallet body position required for the mass measurement experiment (refs. 3, 4, and 5). From these studies, a relatively close approximation can be made for the center of gravity of a shirt sleeved individual seated on the mass measurement pallet. These approximate values are presented in fig. 5. It should be noted that the centers of gravity data exhibit the same general consistency, that is, the mean cg shifts relate as expected to changes in limb position. Thus, as the arms and/or limbs shift, so does the center of gravity. The design of the mass measurement device, therefore, is intended to minimize shifts of the center of gravity. Design features of the MMS that minimize center of gravity shift are: (1) a body restraint harness; (2) pallet foot rests; and (3) pallet back rest.

Table 3

ESTIMATED DESIGN VALUES FOR SELECTED FLIGHT CREW BODY DIMENSIONS AND VALUES

Measurement Description	5th Percentile Basic Nude Dimension (inches)	5th Percentile Shirt Sleeve Dimension (inches)	95th Percentile Basic Nude Dimension (inches)	95th Percentile Shirt Sleeve Dimension (inches)
Sitting Height	33.8	34.1	38.0	38.3
Shoulder (acromial) Sitting	21.3	21.4	25.1	25.2
Buttock-Knee Length, Sitting	21.9	22.0	25.4	25.5
Buttock-Leg Length, Sitting	29.4	39.8	46.1	46.5
Shoulder Breadth, Sitting	16.5	16.6	19.4	19.5
Elbow-to-Elbow Breadth, Sitting	15.2	15.5	19.8	20.1
Hip Breadth, Sitting	12.7	12.8	15.4	15.5
Foot/Sandal Breadth (max)	3.5	4.1	4.1	4.7
Heel/Sandal Breadth	2.40	3.0	2.87	3.47
Foot/Sandal Length	9.8	10.4	11.3	11.9
Weight	132.5	139.9	200.8	208.2

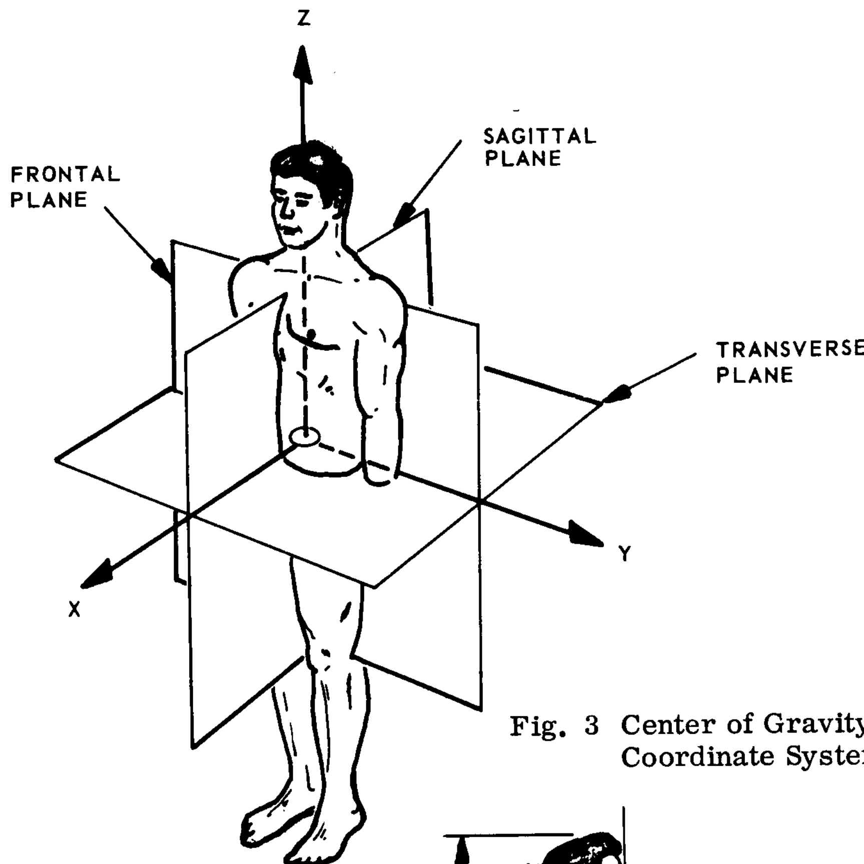


Fig. 3 Center of Gravity Body Coordinate System

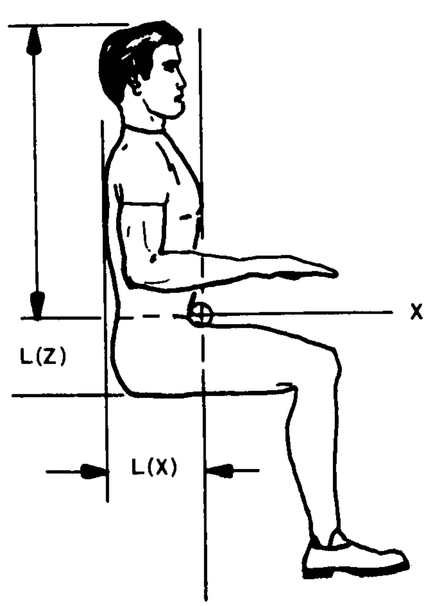


Fig. 4 Reference Landmarks for Location of Center of Gravity

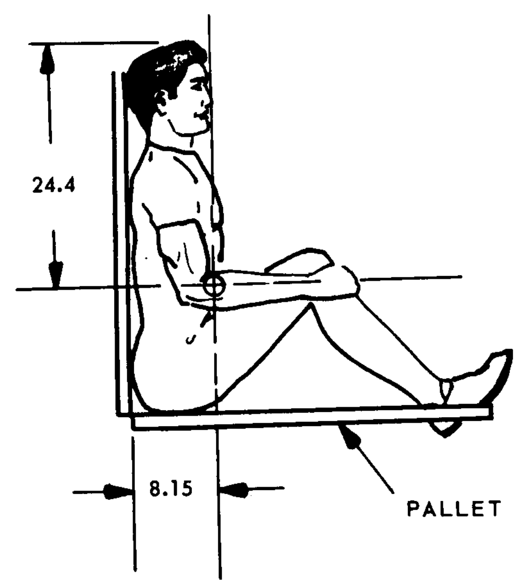


Fig. 5 Pallet Configuration Indicating Approximated Center of Gravity for 50th Percentile Individual

Restraint System

Preliminary testing with the LMSC laboratory model of the MMS indicated the need for a restraint system for both animate and inanimate objects. The animate, or liquid subjects, when oscillated, produced slosh which has an effect on the time period of the complete oscillating spring-mass system. Since the oscillation time is converted directly into pounds, the slosh can affect the mass determination performance of the device. Configuration studies in this area centered on design approaches to a restraint system which would make the mass being determined act as an integral part of the oscillating spring mass system.

Animate restraint system. – Preliminary testing indicated that human subjects acted much like a liquid on the MMS; that is, the oscillating motion would displace internal body organs and fatty tissue which would increase the total period of oscillation thus producing error in the mass determination.

Several restraint systems were studied employing the use of shoulder straps and seat belt. These first studies indicated the need for the restraint system to be integrated with a vest covering the chest and abdomen which could be pulled tight in all directions applying positive pressure to this portion of the body. The sides of the vest must be adjustable to provide a proper fit for people in the 5 to 95 percentile astronaut population. A zipper is fitted to the center of the vest. Shoulder straps, sewn to the vest, pass through the back attachment fittings and on to the parachute quick-release device which locks them in place. The seat belt attaches to the quick release device and the edge of the pallet. Side straps, sewn to the center of the vest, pass through a roller fitting attached to the seat back. When the straps are pulled, load is placed on a series of bungee cords on each side of the vest. This pulls the chest and abdomen in while still allowing some flexibility for breathing comfort (fig. 6).

Quick release from the restraint device is achieved by turning the parachute quick release plate to the unlock position, then pressing inward. This action releases all lock pins and the shoulder harness and seat belt are free from attachment. The vest is then unzipped and the subject is free of the restraint system.

Initial studies indicated that the use of a small elastic head strap would be required for head restraint. However, additional testing with the LMSC laboratory model showed that the test subjects had no problems in holding the head still during the oscillating period of the MMS. Based on this information, and the desire to keep the MMS as simple as possible, this requirement was eliminated.

To determine the mass of liquid objects which are not restrained under pressure or liquids which do not fill the container they are in, presents a problem, especially in the zero-g environment. It was not in the scope of this program to design containers for fluid management in the zero-g environment. However, if we assume that a fluid is contained in a vessel which would reduce slosh and zero-g effects, the design of the MMS should provide a means of holding such a container. It is, however, a design goal to determine if accuracies of ±0.50 lb in the 5-40 lb weight range and ±1.0 lb in the 41-250 lb range can be achieved and the effect of slosh on these accuracies.

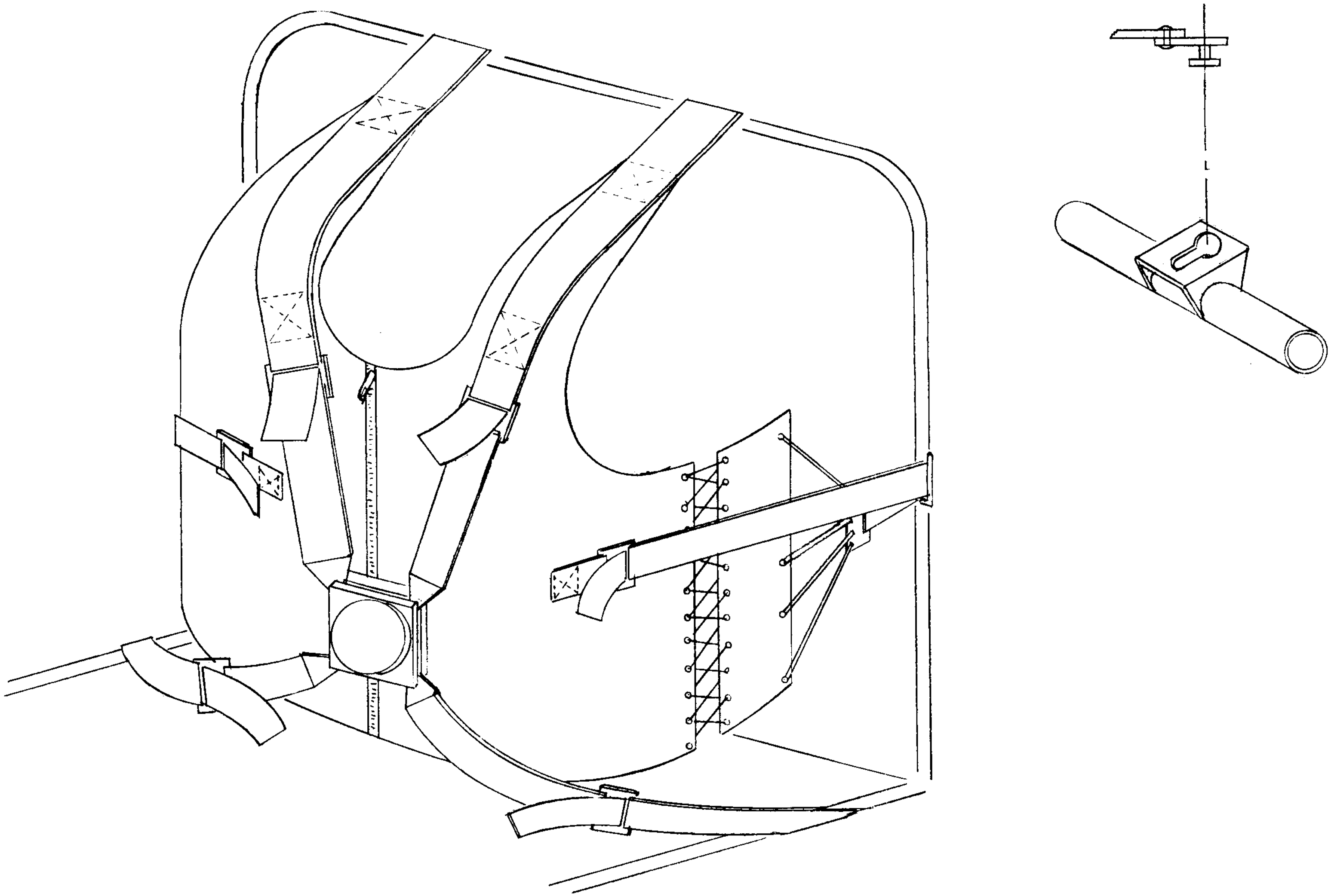


Fig. 6 Man Restraint System

Inanimate restraint system. – If the MMS is to be a versatile tool for determining the mass of a wide variety of objects, a restraint system or clamping device must be designed which would give maximum flexibility to restrain a wide range of objects of many and varied shapes. Studies indicated that the system could best handle small objects in the weight range of 5-40 lb with the seat back in the folded position. Several clamping methods were studied and the relative merits of each analyzed. The restraint system which offered the most flexibility plus positive restraint consists of an aluminum tubular frame attached to pivot fittings of the back surface of the seat back. This feature allows the clamping device when not in use to be folded flat against the back. It is held in place using a spring clip.

The main frame of the clamping device fits inside of the mounted pivot tubes. Three holes through each pivot tube align with three holes in the main clamping frame. Quick release pins lock the main frame and pivot tubes into one structure. Maximum distance between lock pins on the tube assembly is 25.65 in. By removing the quick release pins the main frame can be pulled outwards and locked at three different points to a maximum of 15.2 inches. Mounted to the center of the main frame is a double acme threaded screw fitted with a swivel lock plate. When the restraint system is rotated into position (as shown in fig. 2), the lock plate will align with a rubber pad mounted in the center of the folded back. The object whose mass is to be determined is placed on the rubber pad and the lock plate clamped down on the object in much the same way as clamping an object in a vice. Note that the frame of the clamping device is mounted parallel to the direction of oscillation. This is to prevent independent oscillation of the mass being determined which would effect the timed oscillating period.

To restrain objects of larger size and weight, the seat back is placed in the upright position. The seat back then provides restraint in one direction. To restrain the object in the other plane an aluminum fitting with three bungee cords attached is placed in slots in the edge of the pallet seat. The bungee cords are then placed over the object. The resulting force holds the object to the pallet during the oscillating period. Slots can be placed at various points along the edge of the seat and seat back giving a wide range of attach points and accommodating a wide selection of inanimate objects.

Optimum Mass Measurement System Selection

Prior to start of detail engineering design and fabrication a NASA/LMSC design review meeting was held to present the various designs and optimize the MMS prior to fabrication. Each of the study areas were presented and detail discussions ensued with the following MMS selection.

Threshold acceleration. – The basic requirements for the threshold acceleration system were reviewed in detail to determine if all, or in part could be included in the basic MMS and developed into working hardware. The study indicated that the 150 cm of rail or track is adequate for threshold acceleration studies in the range of 5 to 10 cm/sec^2. However, the study further revealed that many subtleties existed in the area and further investigation must be made before definitive hardware could be designed. The decision was made to fabricate the prototype MMS for mass determination only with further investigations into techniques of determining the otolith sensitivity to very low accelerations.

Pallet assembly. – The honeycomb pallet was selected for further detail design and fabrication. Its selection was based on strength-to-weight ratio, lower system profile and volume, and overall system simplicity.

Carriage assembly. – The carriage assembly, the heart of the MMS, is the most critical subsystem to be designed and fabricated. The reproducibility of oscillation times for any given mass is an absolute requirement. Of the three systems studied, (1) Saginaw ball-bearing spline, (2) Turnomat linear motion bearing, and (3) LMSC multiple conventional ball bearing cluster, two were selected for further design and fabrication. They are the Saginaw ball bearing spline system, and the LMSC conventional ball bearing cluster. This yields two completely different design approaches to providing linear motion. In addition, the square rail or shaft on which the LMSC conventional ball bearing cluster rides is interchangeable with the square Turnomat bearing thus providing a third system as a backup. Both carriage assemblies are interchangeable with the pallet assembly allowing each of these bearing systems to be evaluated during the test phase of the program with relative ease.

Electronic timing device. – Studies indicated that timing accuracy, simplicity and reliability would be required in the MMS electrical system. The electronic system selected to best meet these requirements consists of a light beam being cut by carriage oscillations acting on a photo-sensitive device which in turn sends a signal to a flip-flop circuit which counts a discrete number of pallet cycles and in turn displays this time interval in seconds on a solid state electronic counter. A decision was also made to fabricate a complete breadboard of the system for performance evaluation and operational characteristics prior to fabrication of the end item hardware.

Restraint system. – The study and design of the MMS to weigh objects other than man in the weight range of 5 to 40 lb was explored in detail. The design selected permits the folding of the back of the pallet assembly to form a surface to which a clamping device much like a vice is attached. The clamping device can be raised and lowered to meet the requirements of many different shaped objects. The device is also designed to hold spheres and other spacecraft hardware. Larger objects are held secure with the back in the upright position in conjunction with the use of bungee cords with fittings attached which mate with slots in the edges of the pallet seat and back.

The man-restraint system selected is designed to restrain the torso and to minimize internal organ displacement. The system consists of a parachute-type shoulder harness with lap belt integrated with a nylon knit fabric vest fitted with a zipper.

OTOLITH THRESHOLD MEASUREMENT

This section will concentrate primarily on the problem of measuring otolith organ activity in space. In particular, the results are reported of a preliminary study which evaluated the idea of expanding the use of the mass measurement system for stimulating the otolith organs. Such a use appeared attractive because the controlled, repeatable, in-line harmonic motion of the MMS is the main requisite of otolith investigation.

The widespread but loosely defined concern for vestibular function in weightlessness of a few years ago has become more specific as a result of the information derived from recent space flights and associated laboratory studies. The weightless condition experienced by astronauts thus far has not produced spatial disorientation or clinical deterioration of the vestibule (ref. 6). Researchers are drawing finer distinctions between otolith and canal processes both at anatomical (refs. 7 and 8) and functional (refs. 9, 10, 11) levels. Concurrently, other studies point to the interpendence of the two organs (ref. 12) and to vestibular function interaction with the other sensor modalities (ref. 13).

Present attention of specialists concerned with vestibular function in space appears to be centered on problems related to long-term exposure to the subgravic conditions. These problems can be sorted into two classes. The first entails vestibular disturbances that are traceable to weightlessness per se. The second class includes problems arising from the use of artificial gravity for counteracting the effects of weightlessness.

Most authorities agree that the vestibule will adapt to the weightless condition over the course of a prolonged space flight (refs. 14 and 15). Difficulties are anticipated, however, during the reentry phase of such a flight due to the sharp transition from hypogravic to hypergravic accelerations. The vestibular apparatus will probably not adapt sufficiently during the short interval between gravitational states. Consequently, the astronaut is likely to experience spatial disorientation, postural and visual illusions, or motion sickness to the extent that his performance of critical reentry tasks may be dangerously degreded (refs. 6 and 14).

It appears practical at this time to generate articial gravity either by rotation of the spacecraft or by an onboard centrifuge. The rotational environments produced by these methods may also induce the aforementioned vestibular disturbances thereby vitiating or at least limiting their use as counteractive agents. A number of investigators are exploring this problem area in terms of basic differences in vestibular stresses in various rotational environments, and in particular those environments anticipated in space situations (refs. 16, 17, and 18).

NASA has indicated active concern for vestibular function in space by including two proposed experiments, one involving the otoliths and another the semicircular canals in the Apollo Applications Program (APP), to wit:

Experiment 0101 – Assessment of Changes in Sensitivity of the Otolith Mechanism and Its Central Nervous System (CNS) Connections to Linear Accelerations as a Result of Prolonged Weightlessness.

Experiment 0102 – Effects of Rotation of The Head in Weightlessness and of a Rotating Vehicle on Semicircular Canal Functions.

Otolith Organs

The function of the otolith organs is to sense and transmit neural data to the brain about the position of the head relative to the direction of gravity and about linear accelerations of the head in the horizontal plane. The physical stimulus to the otolith is linear acceleration. The stimulus is detected and transduced by the two sac-like structures in the inner-ear known as the utricule and saccule. Within these structures are found minute calcareous crystals supported by gelatinous material. Theories about the specific mode of stimulation are controversial and less is known about the saccule than the utricule. It appears, however, that when linear acceleration is acting upon the head, hair-like projections of sensory cells embedded within the gelatinous layer are bent by the slight relative displacement of the crystals. This bending of the hair cells produces a transducer effect that results in neural activity in the appropriate, utricular or saccular branch of the vestibular nerve (refs. 19 and 20).

It is an oversimplification, however, to consider the otolith organ strictly in terms of an electro-mechanical transducer. A more complete and rational viewpoint has been advanced in what may be termed a system theory of otolith function. This theory emphasizes the modifying influence upon the patterns of activity in the utricule and saccule by the central nervous system. There is considerable evidence in support of the concept that the vestibular organs, like other sensing mechanisms, are components of closed loop systems under CNS control (ref. 20). A commonly known phenomena in this regard is the habituation to vestibular sensations experienced by pilots, seamen, figure skaters and acrobats. Other evidence is found in objective studies on habituation although these have concentrated largely on the semicircular canals. The work of Grabiel and his associates is notable (refs. 15 and 16). For example, certain subjects living in the Pensacola Slow Rotation room became immuned to motion sickness after several days rotation. Upon cessation of rotation, however, some subjects again suffered motion sickness.

<u>Intradependence of vestibular function</u>. – A number of writers have underscored the strong reciprocal relationship of the otoliths and semicircular canals. Commenting on the characteristics of motion sickness, Lansberg (ref. 21,) points to the continuing controversy over whether it is caused basically by the otoliths or the semicircular canals. He observes that the "cause" cannot be attributed to either organ exclusively: "Passed are the good old days when the semicircular canals and otoliths were separate organs with separate functions and separate effects" (ref. 21, p. 71). In support of this statement, he presents the results of experiments indicating

that the otoliths can modulate an ampular signal and have a nystagmus-modifying or even nystagmus-generating power.

Along similar lines are the experiments performed on the Coriolis Acceleration Platform at the U. S. Naval School of Aviation Medicine, Pensacola (ref. 12). Subjects were exposed to sinusoidal linear accelerations at frequencies of 0.2, 0.4, and 0.8 cycles per second, peaking at 0.6 g in the horizontal direction. When the subjects were oriented such that the stimulus acted either in the horizontal or frontal planes a sinusoidally varying nystagmus was produced. These results for the first time show that systematic horizontal nystagmus can be elicited by linear accelerations. The authors are careful to point out that they do not claim that the responses issue exclusively from the otoliths or the canals. Possibly an even more significant example of intradependence of vestibule function is the finding in experiments by Sasaki, et al (ref. 22), that the otoliths of the right and left inner ears suppress and augment each other's activities.

Vestibule interdependence with other senses. – The perception of body position is a complex response involving not only the vestibular organs but the other sensing modalities as well. The interaction of sight and sound sensors, skin, muscle, and deep body sensors, has been termed cross-modality interaction by Clark (ref. 13). The oculogravic illusion, a much-used test of otolith function, is influenced by postural reflexes, and other sensory phenomena (ref. 23). For instance, the threshold of the illusion varies as different visual cues are presented to the subject. Also, recent investigations have shown that a lag in the perception of the oculogravic illusion (inertial-lag effect) ensures from a change in the direction and magnitude of resultant force acting on a subject (ref. 24).

Otolith stimulation and responses. – The otoliths can be stimulated either by linear acceleration of the subject's body or by tilting his head in the gravitational field. Commonly used devices for generating these stimuli are parallel swings (ref. 11, 25, and 26), vehicles propelled over linear tracks (ref. 12), and centrifuges rotated at constant angular velocity to produce centrifugal acceleration (refs. 11 and 27). An intriguing modification of the centrifugal acceleration principle is a device described by Johnson (ref. 28). The device consists of a main turntable upon which is mounted a counter-rotating secondary turntable. The writer claims that the device does not introduce angular acceleration and hence does not stimulate the canals.

For the measurement of otolith responses, subjective methods seem to predominate. The subject indicates his reaction by an oral or gestural communication and/or the experimenter observes some outer physical manifestation of the reaction. Typical subjective responses are changes in spatial orientation such as the oculogravic illusion, somatic inversion, or uprightness, and other body sensations experienced by the subject when exposed to linear acceleration at an angle to gravity.

Somewhat more objective is the measurement of human otolith response as reflected by ocular counter-rolling. When a subject's head is inclined laterally, his eyes perform a rolling movement opposite to the direction of inclination. Miller (ref. 29) gives a short history of ocular counter-rolling experimentation, and describes a photographic technique in which eye rotation is measured to an accuracy of

±5.3 minutes of arc. An important milestone in the evolution of this technique was established by recent experiments in which otolith organ responses were measured in earth standard, one-half standard, and zero gravity environments (ref. 30). The latter two conditions were produced by parabolic flight maneuvers. Another method that may prove to be more objective as well as more selective than the ocular counter-rolling method has been reported by Sasaki, et al (ref. 22). These researchers employed electromyograms of the anterior tibial muscle to record thresholds of the spike discharge while linear accelerations of various magnitudes were applied to the subject. The thresholds determined this way were comparable to those determined by means of subjective responses, i.e., about 9 cm/sec^2.

Parallel Swings

Among the devices described above for imparting linear acceleration to humans, the parallel swing is the most pertinent to the purposes of this report. A parallel swing consists essentially of a platform suspended by four cables. The platform remains parallel to the floor even though it describes an arc as it swings back and forth. Thus no angular accelerations of levels that might excite the canals are imposed on the subject.

When released from some point above its resting position, the platform is driven by gravity. The force of gravity is resolved into two components, one tangent to the arc of motion equal in magnitude to mg sin θ and the other proportional to the tension on the suspension cables equal in magnitude to mg cos θ , as shown in fig. 7. Van Egmond, Groen, and Jongkees (ref. 11) point out that by mechanical analysis the direction and magnitude of otolith displacement would appear to be dependent upon the resultant of the vector of gravity and the vector representing the otolith reaction to tangential acceleration.

This resultant is mg cos θ (fig. 7). At the return points of the swinging motion, this resultant attains its greatest angular displacement relative to gravity. Since the platform maintains a constant orientation relation to gravity the subject would experience a tilting sensation in the direction of motion. In studies testing this hypothesis, the authors reported that the subjects experienced both linear movement and tilt. They ascribed the sensation of linear movement to the skin senses and the sensation of tilt to the otoliths. In fact, they were so certain that tactile stimuli were the principal product of the parallel swing they felt that this device could not be used for measuring otolith thresholds to linear movement accurately. The authors decided that this could best be accomplished by placing the subject in a rotational chair and rotating him at a constant angular velocity. The centrifugal acceleration thus engendered would be in effect a linear acceleration without the artifacts produced by the to-and-fro movement of the swing.

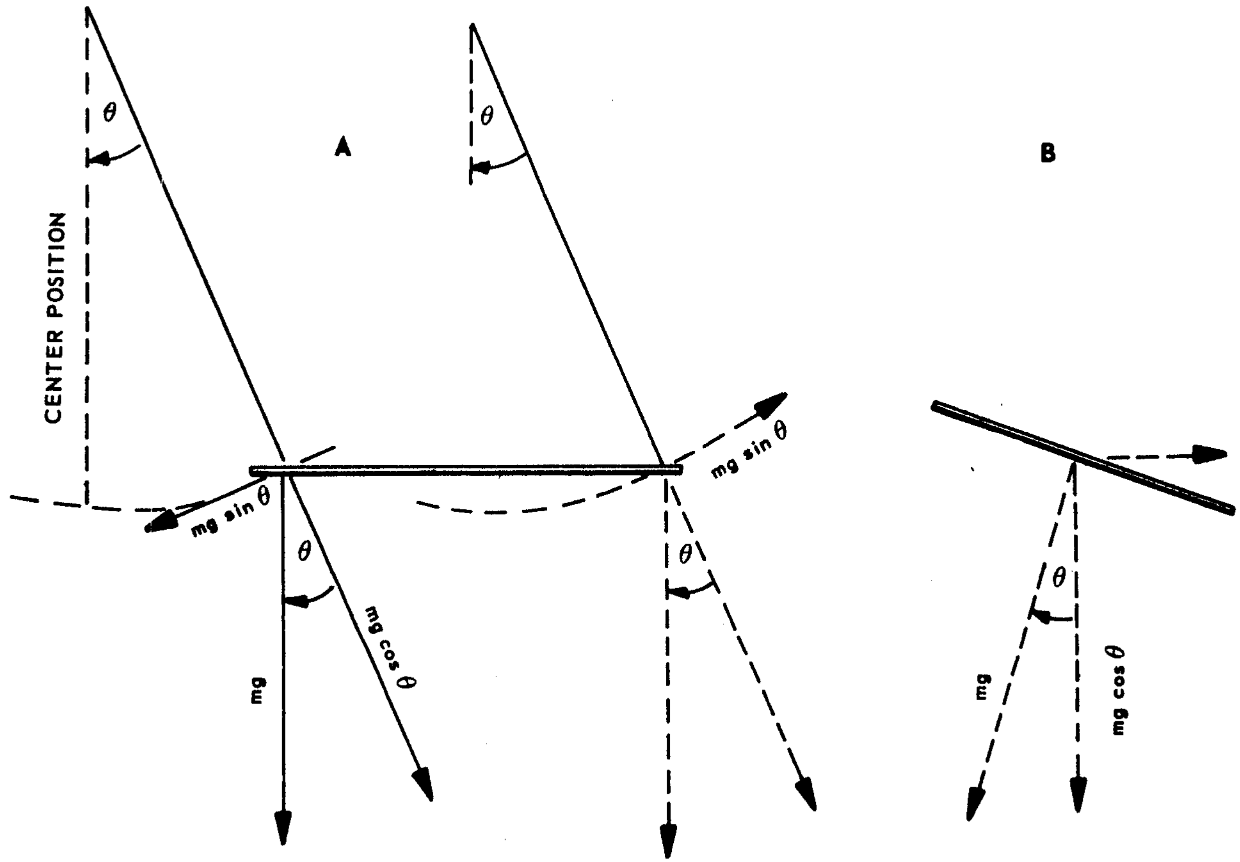


Fig. 7 Parallel Swing Geometry (modified from ref. 11)

Illustrating the kinetics of the parallel swing and the "inertial force" which would influence otolith displacement. Dotted arrows are the "inertial vectors." In B, the resultant, mg cos θ, is rotated to correspond to the usual pictorial representation of gravity. Centripetal acceleration, which would be absent at the turning point, is not included in this representation.

Guedry and Harris (ref. 31) question the conclusions of Van Egmond, Groen and Jongkees. They cite Walsh's observation, namely, that perception of linear movement is primarily an otolith function since it is elicited by stimuli of lesser magnitude than is required for perception of tilt. Further, they suggest that Van Egmond and his colleagues base their analysis on the assumption that the instantaneous resultant force represented by the rotating vector would be accurately sensed by the subject. This assumption is not necessarily valid since the subject's tilt sensors might not react quickly enough to the rapid angular excursion of mg cos θ (fig. 7). To test this contention Guedry and Harris employed normal subjects and labyrinth defective subjects (L-D subjects) to compare their responses in various motion modes of a parallel swing. Essentially, they found that none of the twelve normal subjects perceived tilt, and that all perceived lateral oscillation in the horizontal plane. On the other hand, six of the eight L-D subjects reported apparent tilt sensations. Two of the L-D subjects did not perceive linear velocity or displacement. The authors point out that these results are in direct conflict with the claim of Van Egmond, et al (ref. 11), that the parallel swing elicits principally the sensation of linear velocity from the extra-labyrinthine sensors, and principally the sensation of tilt from the otoliths. They conclude that the parallel swing appears to be a valid tester of otolith function and that additional experimentation with the device is warranted.

Trade-off analysis. – A preliminary engineering system analysis was performed to determine the most suitable means of imparting linear acceleration to astronauts. This took the form of an independent trade-off analysis of the various ways of producing linear acceleration in space (Appendix A). A summary of the analysis is given in Table 4. The table indicates that the harmonic motion principle is one of the better approaches to the problem.

Conclusions of background study. – As a result of the literature survey and the trade-off analysis it was concluded that the harmonic motion principle constitutes a valid and practical technique for stimulating the otolith organs. It is recognized that careful consideration must be given to the note of caution sounded by researchers, i.e., that even with painstaking experimental controls it cannot be assumed that the sensation of motion elicited by the oscillating platform is exclusively due to otolith reaction.

Adaptation of Harmonic Motion Principle to Space Conditions

The harmonic motion of an oscillating platform is produced by gravity hence the method is not directly adaptable to the space environment. It was suggested by Dr. Miller, in a communication to LMSC, that the Grabiel-Miller Litter Chair (GMLC) could be modified for use as an otolith stimulator in subgravic conditions. This could be done by using the GMLC in its in-line configuration and driving it in harmonic motion by mass-spring action. In substance, this is a description of the mass measurement system. Since the latter is being developed specifically for space vehicles, it would appear the additional design effort for expanding its function to include otolith testing would be relatively minor. Another factor in favor of the MMS in contrast with typical parallel swings or oscillating platforms is that it more closely approaches the objective of true, reproducible, in-line motion.

Table 4

COMPARISON OF METHODS OF GENERATING LINEAR ACCELERATION

Characteristics / Types of Motion	Critical Problems	Critical Measurements* for Determining Acceleration	Drive Equipment Required for Achieving Various Acceleration Rates	Compatibility With Mass Measurement System
Acceleration in Rectilinear Motion	1) Varying Acceleration rates and force – source 2) Displacement length	1) Acceleration directly or 2) Measure S & t continuously	Servo Motor	Different test profile
Uniformly Accelerated Rectilinear Motion	1) Constant force source 2) Displacement length	1) Acceleration directly or 2) Measure S & t	Servo Motor or range of constant force Negator Springs	Different test profile
Acceleration in Curvilinear Motion (Parallel Swing)	1) Increased volume requirement 2) Varying acceleration rates 3) Acceleration meas. 4) Curved track required	Measure V & r for a_n Measure S & t for a_t Solve $\sqrt{a_n^2 + a_t^2}$	Servo Motor or Springs	Different test profile
Simple Harmonic Motion	1) Varying acceleration rates	1) Acceleration directly or 2) Amplitude and frequency	Springs having various spring constants	Identical test profile
Circular Motion (Centrifuge)	1) Closed circular device required	1) Angular velocity and radius	Servo Motor	Different test profile

*Note: Types of motion are discussed in Appendix A.

Preliminary Tests

An exploratory study was conducted for insight into problems connected with the measurement of otolith function, and implementation of the harmonic motion principle. A specific aim was to determine if typical otolith thresholds could be obtained from stimuli produced by a spring-mass system. Corollary objectives of the study were the following:

- To become familiar with otolith measurement procedure such as selection of subjects, methods of otolith stimulation and recording of thresholds, and control of experimental variables
- To assess the practicality of producing linear acceleration by a short-track, oscillating platform driven by springs
- To determine spring design criteria for obtaining linear acceleration ranging from 5 to 10 cm/sec^2

Subjects. – The subjects were four males selected from the staff of Biotechnology Organization. Selection was based on their passing the Romberg postural equilibrium tests, and the Grabiel-Fregly ataxia test battery (ref. 32). The best three out of five trials were used in scoring. A short description of the test follows:

(1) Classical Romberg Test: – Standing with eyes closed for 60 seconds, arms at sides and feet together on the floor.

(2) Sharpened Romberg Test: – Standing on the floor with eyes closed for 60 seconds in the stringent body position of body erect or nearly erect, arms folded against chest, feet tandemly aligned in heel-to-toe position.

(3) Ataxia test battery (short version): – Stringent body position, arms folded against chest and performing the following maneuvers:

(a) Walking with eyes open on a 3/4 in. wide rail, 8 feet in length, in five steps.

(b) Standing with eyes open for 60 seconds on the 3/4 in. wide rail.

(c) Standing with eyes closed for 30 seconds on a 2-1/4 in. wide rail.

(d) Standing on one leg with eyes closed for 30 seconds.

Equipment. – A laboratory model was constructed of an oscillating chair (fig. 8). The model consists of a seat and backrest constructed of two pieces of plywood, a spring mechanism, and four suspension cables 35 feet in length. A Unistrut channel is used for spring tie-down and for setting spring tensions quickly and easily. Turnbuckles permit vertical adjustment of the unit for alignment with spring motion. To control visual and auditory cues a helmet is provided equipped with both clear and opaque visors, ear muffs, and a two-way communication system.

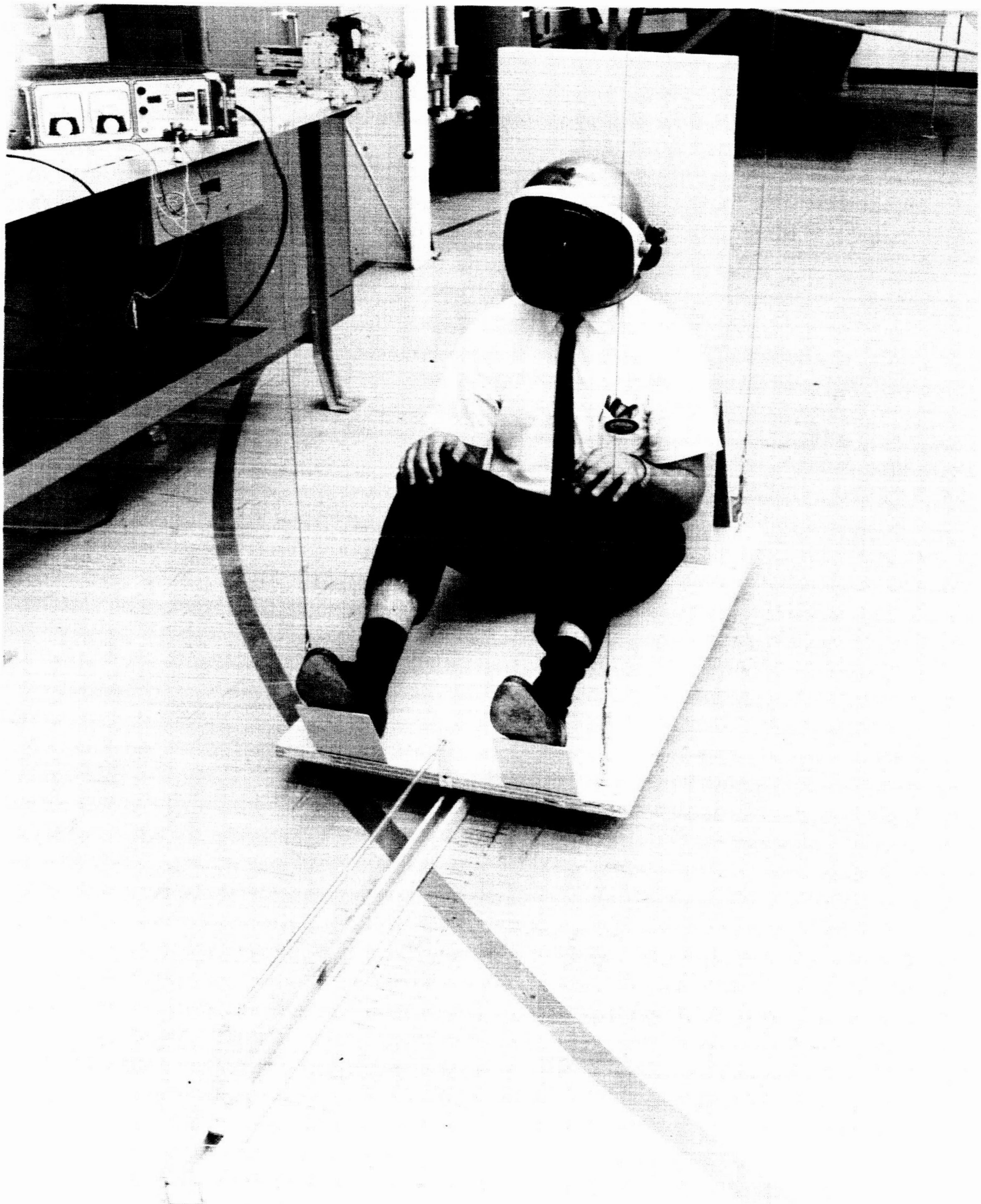


Fig. 8 Laboratory Model of an Oscillating Chair Showing Subject Wearing Helmet to Control Visual and Audio Cues

The unit was calibrated with a ±0.25 g accelerometer, (Systron-Donner Model 4310) lead weights, a voltmeter, and a power supply. The Systron-Donner calibration curve (acceleration calibration vs DC volts) is reproduced in fig. 9. Calibration runs using weights of 45 lbs and 245 lbs established the relationship between the DC voltage output of the accelerometer and displacement of the chair from its resting position. This relationship is shown in fig. 10. Corresponding acceleration values are given in fig. 11. Figure 12 shows the relationship between acceleration and weight for various displacements.

Procedure. — A principal aim of this study was to determine if typical otolith thresholds could be measured from subjects in simple harmonic motion produced by the energy interchange of a spring-mass system. To accomplish this aim the following test procedure was employed:

(1) Subject assumes a sitting position in the test device with head tilted forward approximately 5°.

(2) Acceleration stimuli are produced by an initial 0.5 in. displacement of the device and allowing it to oscillate three times, followed by successive 0.5 in. increments of displacement and corresponding oscillations until threshold is reached.

(3) The subject reports the following responses: onset of moving sensation, direction of motion, offset of moving sensation, lateral motion, if any, and other sensations he cares to report.

(4) Subjects are given 30 training trials.

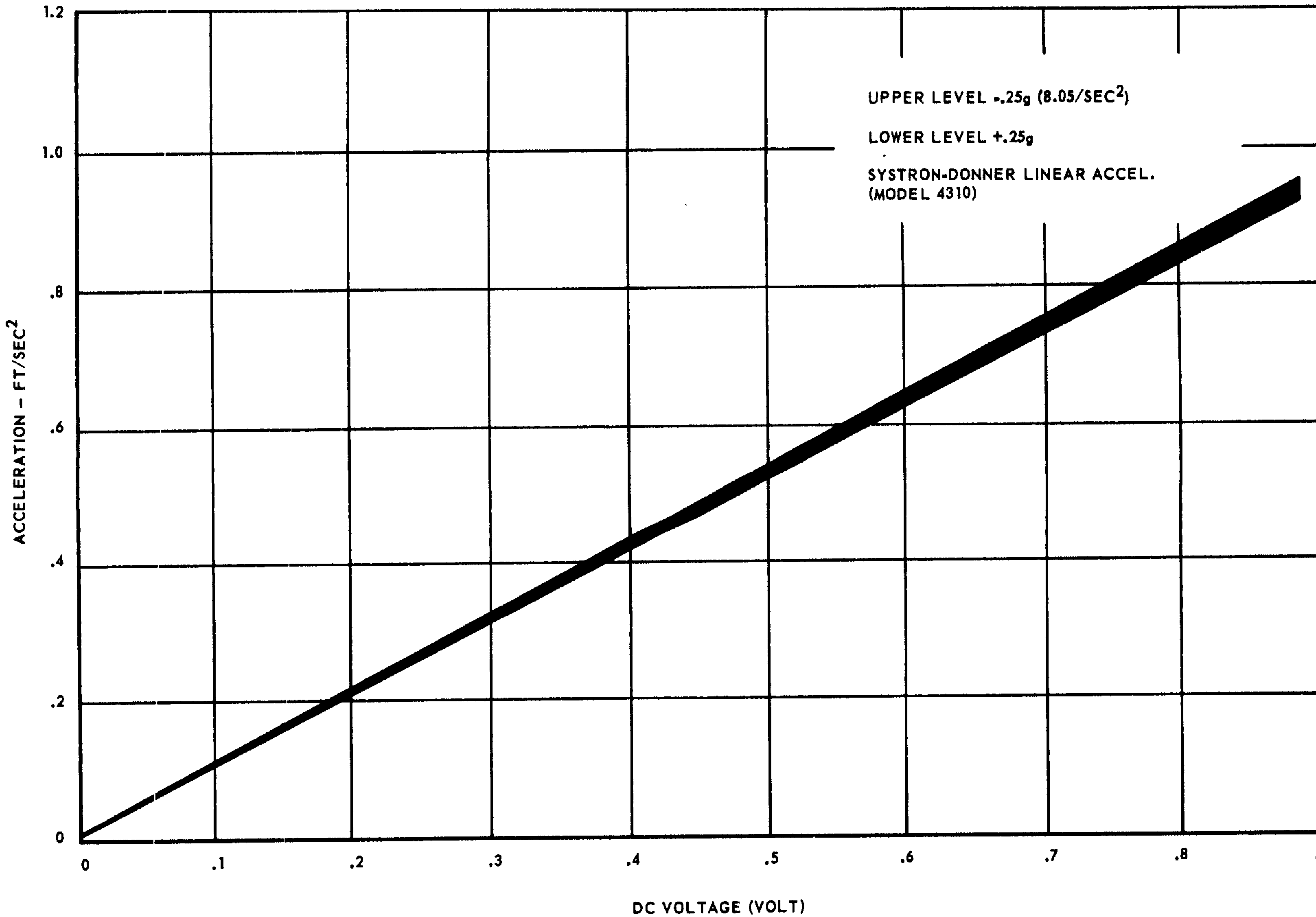


Fig. 9 Acceleration Calibration Showing Relationship Between Acceleration and DC Voltage

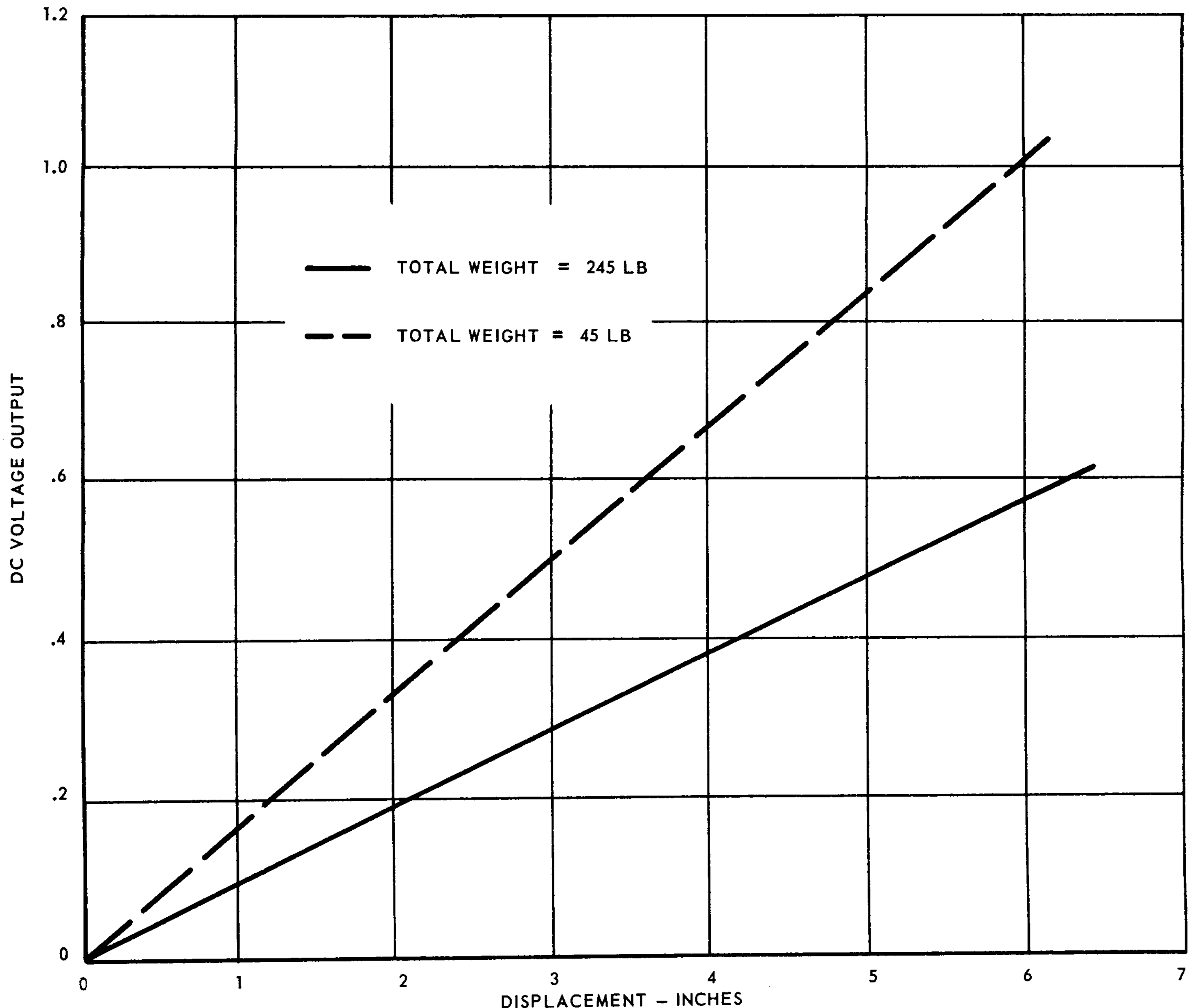


Fig. 10 Relationship Between DC Voltage and Displacement

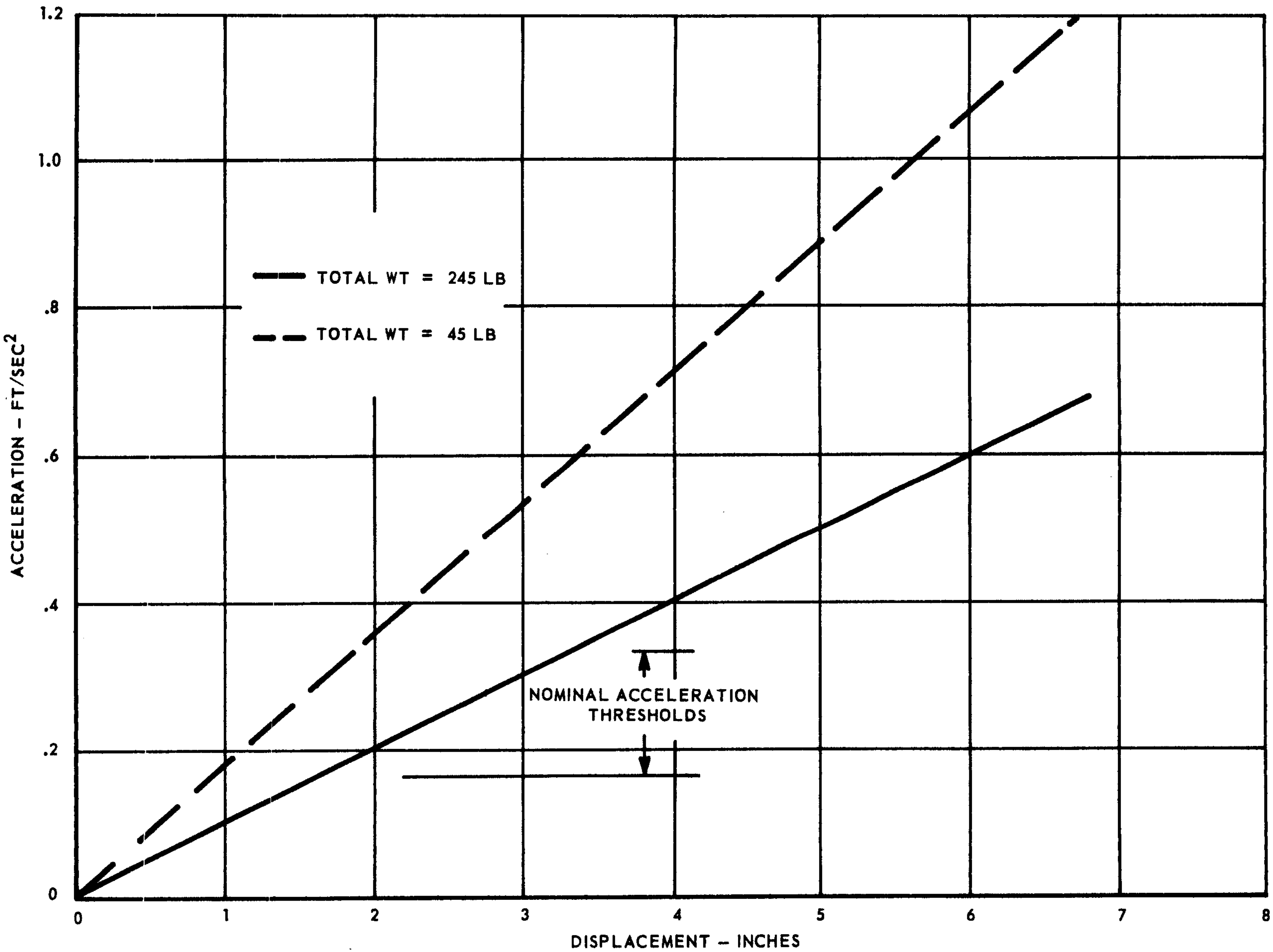


Fig. 11 Relationship Between Acceleration and Displacement

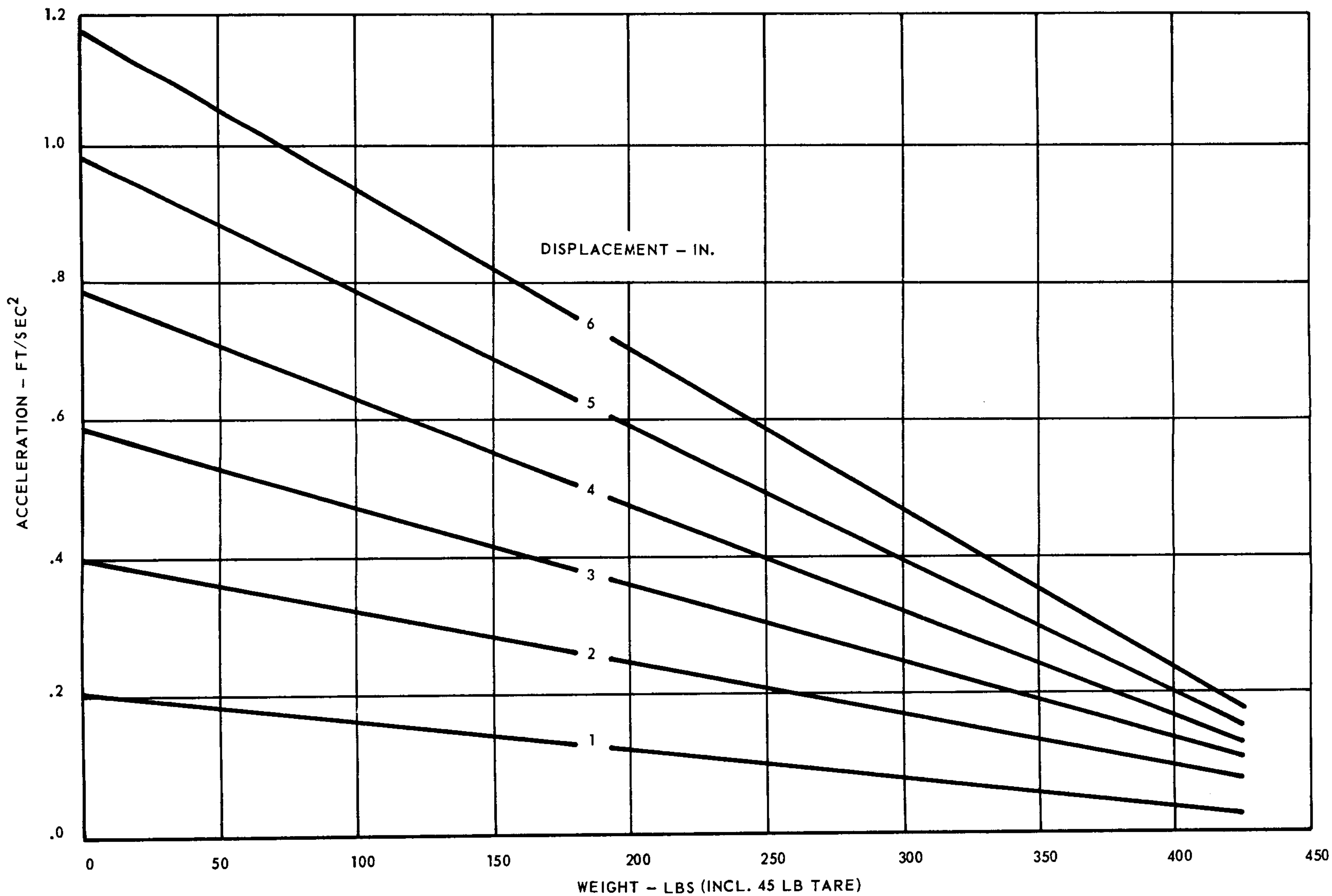


Fig. 12 Relationship Between Acceleration and Weight for Various Displacements

Discussion and Recommendations

The test results and experience gained from the preliminary study suggest that the MMS, appropriately modified, could become a valuable aid in the monitoring of otolith function in space. Before initiation of the design effort, however, additional laboratory studies are recommended. These would methodically examine the problem area outlined below.

Experimental. – The effect on otolith thresholds of the following experimental conditions:

- Head attitudes relative to the direction of acceleration
- Combinations of amplitude and frequency
- Rates of onset and offset of acceleration
- Seat padding methods
- Body constraint methods
- Stimulus generating devices; i.e., parallel swing, oscillating platform, and the MMS
- Response recording techniques; i.e., electrooculograms to determine if nystagmus occurs at low acceleration, and electromyograms of the anterior tibial muscle (ref. 22).

Engineering. – Further work is called for in spring selection. Of particular interest is the development of "soft" springs to obtain longer displacements. Another area requiring special attention entails the design of hardware that produces in-line motion without distracting cues such as noise, vibration, and rough or discontinuous motion.

DETAIL ENGINEERING DESIGN & FABRICATION

Upon completion of MMS optimum design selection, the program moved into the final detail engineering design and hardware fabrication. This task was accomplished in two basic engineering releases. First release was the engineering for the pallet assembly followed by the release of the carriage and electronic subassemblies. While some changes were made, the designs proceeded as defined in the configuration studies. The following paragraphs will outline subsystem development details.

Pallet

Detail engineering of the pallet was accomplished without significant problems. The fabrication of the sheet metal parts to this subassembly are straightforward and no production problems were encountered. Bonding was accomplished using sheet type adhesives which are cut to shape. The adhesive sheets are placed between the outer skins and the honeycomb. The assembly is then vacuum bagged and placed in an oven. The heat melts the adhesive between the skins and the honeycomb while atmospheric pressure forces the assembly tightly together. Proper tooling is required during this operation to prevent slippage of parts relative to each other. Some problems were experienced in this area. The pallet back outer edge channels moved inboard during the bonding process causing a small void between the edge of the skins and the heel of the channel. This void was filled with epoxy and machined smooth after the proper epoxy cure time.

To reduce overall system weight, the pallet skin panels are 0.012 in. sheet aluminum. This very thin sheet is very susceptible to marking and dents produced by normal shop handling and usage. While the thickness used is structurally sound and should be used for flight hardware, some advantages can be seen for using a minimum of 0.025 skin for prototype and test hardware to prevent handling and usage damage. The completed pallet can be seen in fig. 13.

A design improvement was realized in the fitting connecting the pallet seat to the seat back. A quick-release pin located in each attach plate and passing into the seat back secures the back in an upright position. Upon completion and final assembly it was noted that relative movement between the back and seat existed. This movement would cause the mass being determined to move, thus again affecting the timed oscillating period. To correct this problem, a stainless steel bushing was machined to match the diameter of the purchased quick release pins. The bushing was then pressed into each aluminum attach plate, then fitted to the pallet seat and secured in place.

Carriage

Since the Saginaw linear ball bearing and the conventional ball bearing designs each had certain advantages, it was decided to build both, using common components wherever feasible. Common components are employed in the latching and release mechanism, the mounting ring (for mounting the pallet to the carriage, as well as attaching the inner ends of the springs) and the springs themselves. The design of the springs, their

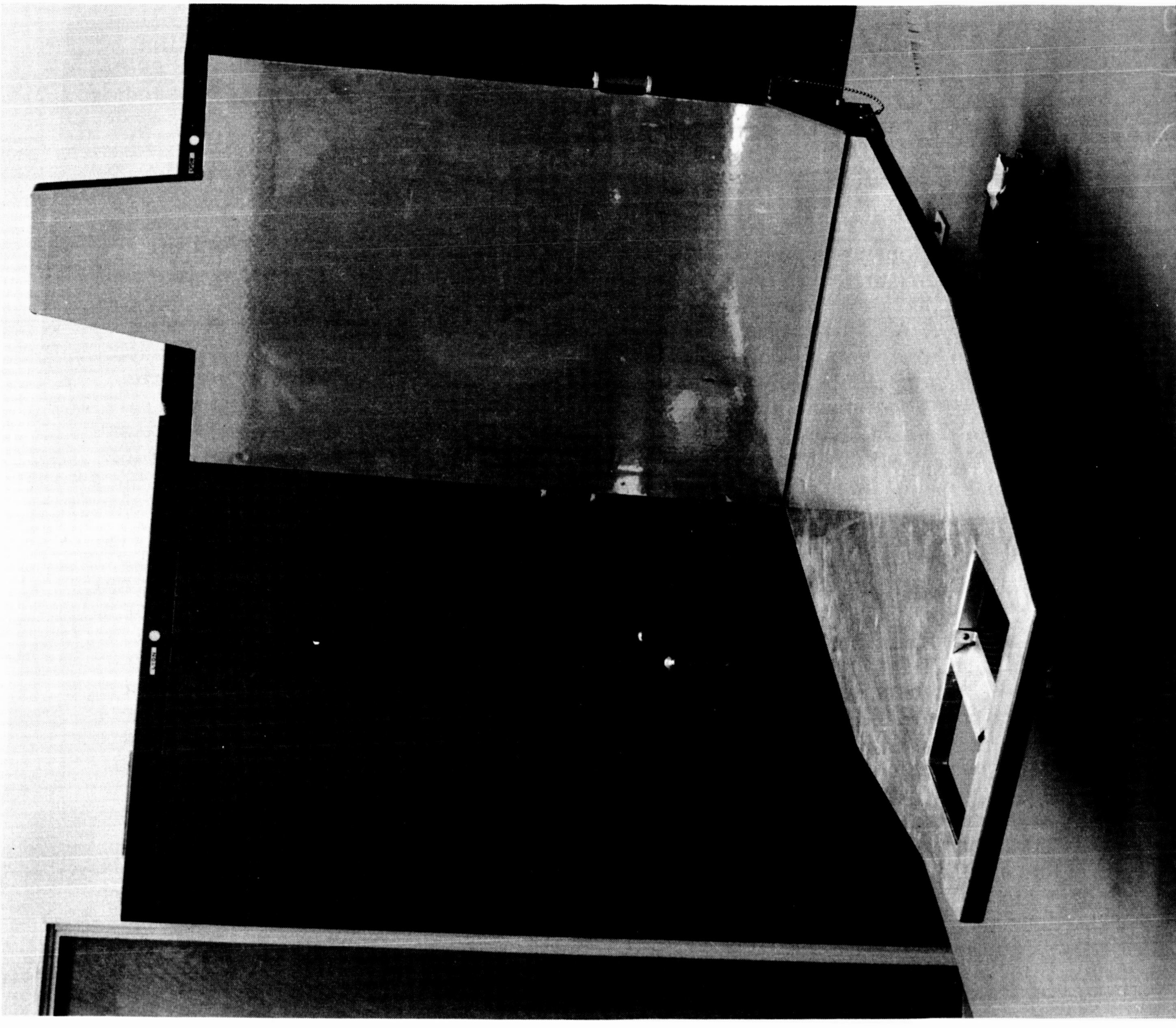


Fig. 13 Pallet Honeycomb Sandwich Panel

mounting hooks, and the latching mechanism provides a great amount of flexibility. For example, the various parameters were chosen to allow any cocking force up to 66 pounds to be used without overstressing the springs. Of course, the initial stretch of each spring (with the carriage centered) must be greater than the subsequent travel of the carriage from its centered position to either extreme. Otherwise, the springs would become completely relaxed during the extreme points of travel, and this condition would not conform to simple harmonic motion.

The conventional LMSC ball bearing design, shown in fig. 14, employs eight Fanfir double-row, double-sealed ball bearings. Double-row ball bearings were chosen to withstand overturning moment without binding. Double-sealed bearings were selected to keep out dirt and dust as well as to retain a light lubricant (approximately SAE 10). General Electric Versalube F 50 was used since it is a "spaceworthy" lubricant. Four of the bearings are mounted on an ordinary straight axle shaft, while the other four are mounted on eccentric axle shafts. Consequently, these latter four bearings can be moved inward or outward by rotating the eccentric shafts on which they are mounted. This provides adjustment to fit the square shaft on which they ride. To prevent binding, there must be a small clearance. However, such a clearance would mean that some of the ball bearings would be in contact with the shaft only intermittently, and would, therefore, be rotating intermittently. This would impose an intermittent inertia, which would cause erratic results. To prevent such undesirable effects, a groove was cut into the outer race of each ball bearing and a silicone rubber O ring was installed in each groove. The groove was shallow enough to allow each O ring to protrude beyond the outer diameter of the bearing by approximately .015 in. This ensures that all of the ball bearings will be rolling in unison. A similar problem could exist between the bearing axle shaft and the inner race of the bearing. To lock these two elements together a single knurl was raised on the axle shaft, providing for an interference push fit.

Except for the bearings, shafts, fasteners, and certain latch components, nearly all of the remaining carriage parts and shaft supports are made of aluminum. The carriage on which the conventional ball bearings are mounted is a built-up welded assembly in which the axle shaft holes were machined after completion of welding to eliminate misalignments due to welding warpage.

One of the noteworthy features of the Saginaw bearing carriage involves the protective bellows. Since the air in the bellows at each end would be alternately compressed and expanded, causing undesirable damping of the carriage oscillations, a pair of bypass hoses was provided to connect the two bellows. These bypass hoses, together with an ample annular clearance between the bellows inside diameter and the Saginaw shaft outside diameter, enables a free flow of air from end to end. Figure 15 shows the complete assembly.

Electronic Hardware

The input to the electronic timing device is provided by the interruption of a light path as the pallet cycles on its track. The first impulse triggers the Hewlett-Packard counter and initiates the count through the pulser. After the desired number of cycles, a signal from the pulser gate stops the counter. Thus, the counter reads out the time for the pallet to make a discrete number of cycles.

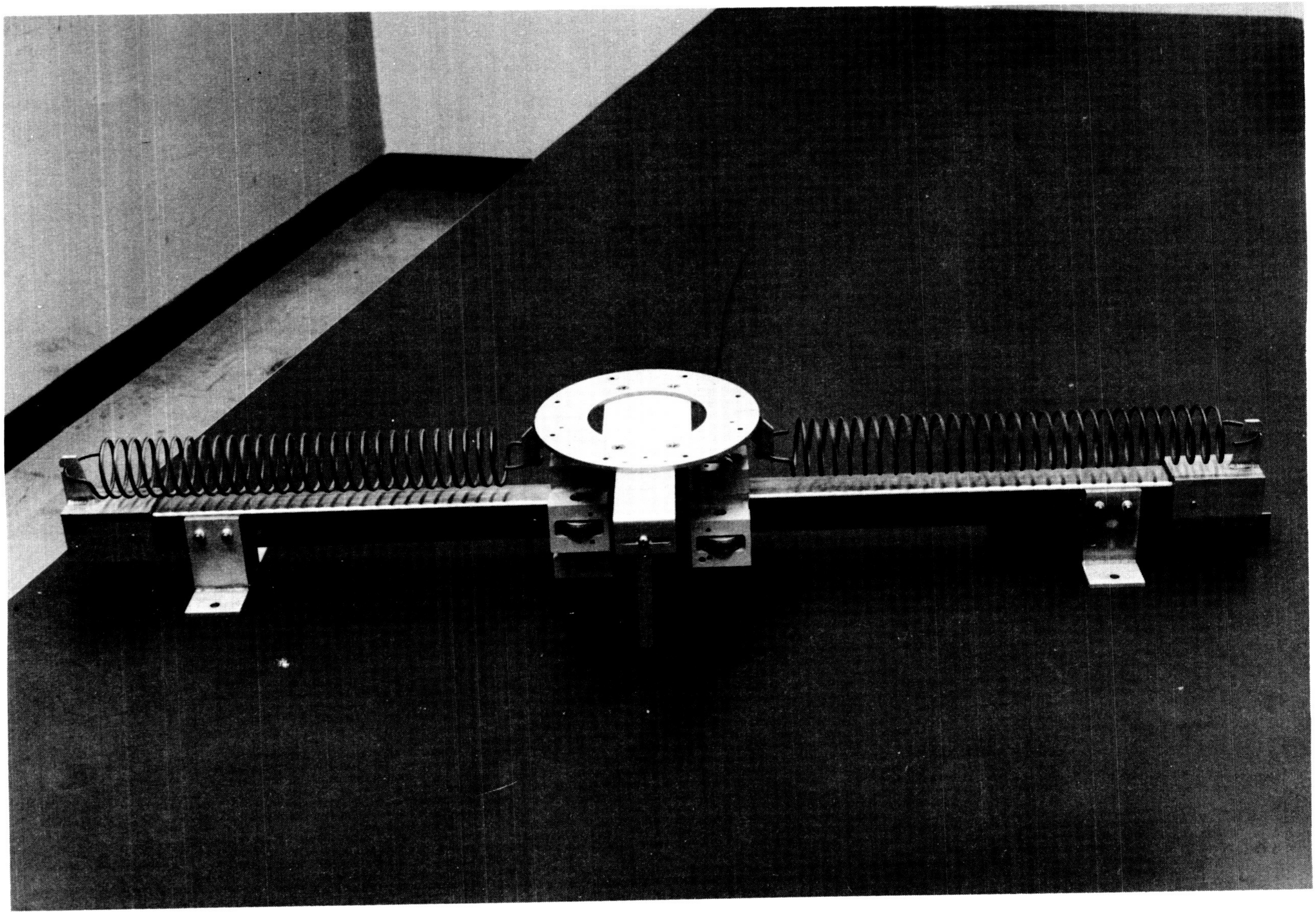


Fig. 14 LMSC Multiple Conventional Ball Bearing Cluster Carriage Assembly

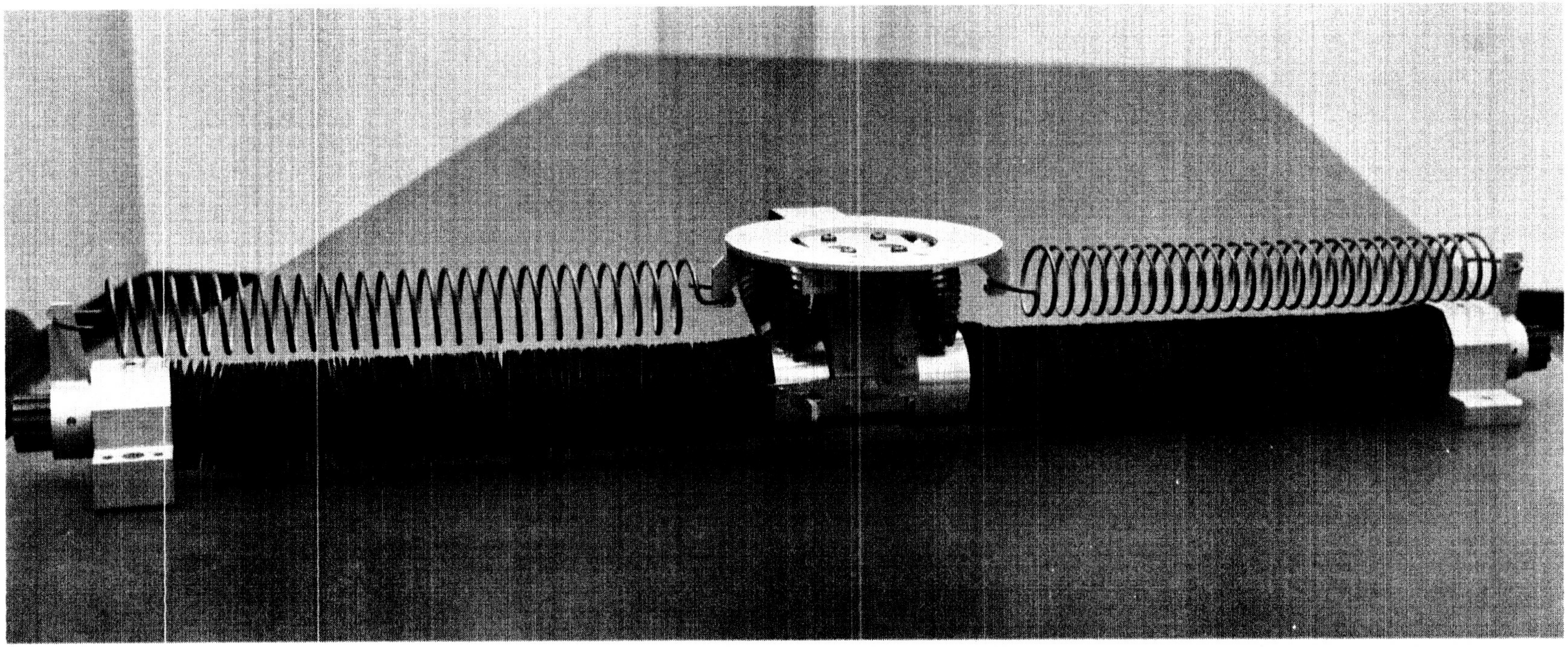


Fig. 15 Saginaw Linear Ball Bearing Carriage Assembly

To sense the light interruptions, a light-sensitive transistor is used in series with a resistor. The collector-emitter junction drops to about 0 volts when the light reaches the base and increases to about 4.5 volts when the pallet cuts the light path. Therefore, as the pallet motion breaks the light path, the transistor output varies from 0 to 4.5 volts.

The pulser uses this pulse variation to count the number of cycles. Four bistable multivibrators are connected as a ripple counter with the first section pulsed by the light sensitive transistor. One output from each of the multivibrators is fed to an "AND gate" which provides an output at the end of the desired number of cycles. The preset number of cycles can be changed by properly selecting the gate inputs.

The Hewlett-Packard counter is started by the first pulse and stopped by the gate output, thus providing the desired time interval.

Timing error is made up of contributions from the pulser and the Hewlett-Packard counter. Table 5 shows the error breakdown.

Table 5

ELECTRONIC TIMING DEVICE ERROR (for one measurement)

Pulser	
Multivibrators	640×10^{-9} sec
Gate	120×10^{-9} sec
Light Sensitive Transistor	30×10^{-6} sec
Hewlett-Packard Counter	
±1 Count	1×10^{-4} sec (at .1 ms setting)
Trigger error	14×10^{-10} sec
Timebase error (aging)	±2 parts in 10^6/week

Total System Error = 1×10^{-4} sec (due to ±1 count of H. P. Counter)

The pulser has been designed for minimum size and weight while permitting ease in disassembly for test and parts replacement. All electronic components used for the pulser are integrated circuit assemblies packaged in a TO-5 container. These components are very small devices for the number of components involved, easily obtainable and relatively inexpensive when compared with conventional components. The light sensitive transistor package was chosen because of the ease with which it can be assembled and its small size.

The pulser is made up of a small circuit board and a panel light source. (See fig. 16.) The board holds the multivibrator, gate and light sensitive transistor. When placed in the square housing, the sensitive end of the photo-transistor faces the panel light beam with a separation of about 3/8 in. When the light path that is created by this separation is interrupted by the pallet motion, a fast pulse is generated by the sensor.

Replacement or test of the pulser board is accomplished by removing the connector plate, at the end of the housing holder, and unsoldering the panel light leads. The board can then be completely removed exposing all components and interconnections for test or a replacement board can be inserted in the housing.

The power supply consists of an Lectrotech unit supplying 28 volts. This is further stepped down to 4.5 volts, for the pulser, by a resistor divider network. A zener diode is used to prevent damage to the pulser should the main supply voltage become uncontrollable. The complete integrated assembly is illustrated in fig. 17.

Restraint System

The detail design of the clamping device paralleled the design developed during the configuration studies. The clamping frame is pinned to fittings which are fastened to the back surface of the pallet back. Solid aluminum blocks are bonded between the skins of the honeycomb structure to receive fittings and distribute the clamping loads. Rubber pads are provided on the pallet back as well as the clamping screw. The pads are designed to prevent slippage between the clamping device and the item being clamped. No fabrication problems were encountered. The clamping device can be seen in fig. 18.

The man restraint system was fabricated per the design evolved during the configuration studies by Security Parachute Company, San Leandro, California. A mockup of the pallet seat and back was fabricated of plywood and used as a base from which to develop the patterns required for fabrication of the final restraint system. Various individuals whose bodly dimensions fell within the 5th and 95th percentile range were seated in the mockup and measurements taken for shoulder strap placement, cut and shape of the vest, zipper placement, abdominal restraining bungee strap placement, side take-up laces and seat belt placement. From these measurements a prototype was fabricated and the system checked for fit and function using human subjects. Subjects' comments as to strap location and degree of restraint were noted and changes and improvements in the design ensued. The final design of the prototype was approved by LMSC and the final end item hardware was fabricated. Upon receipt of the hardware at LMSC further interface checks were made with the pallet and found to be satisfactory. Ease of attachment to the pallet was demonstrated as well as subject quick release from the restraint system. Figure 19 shows the completed assembly.

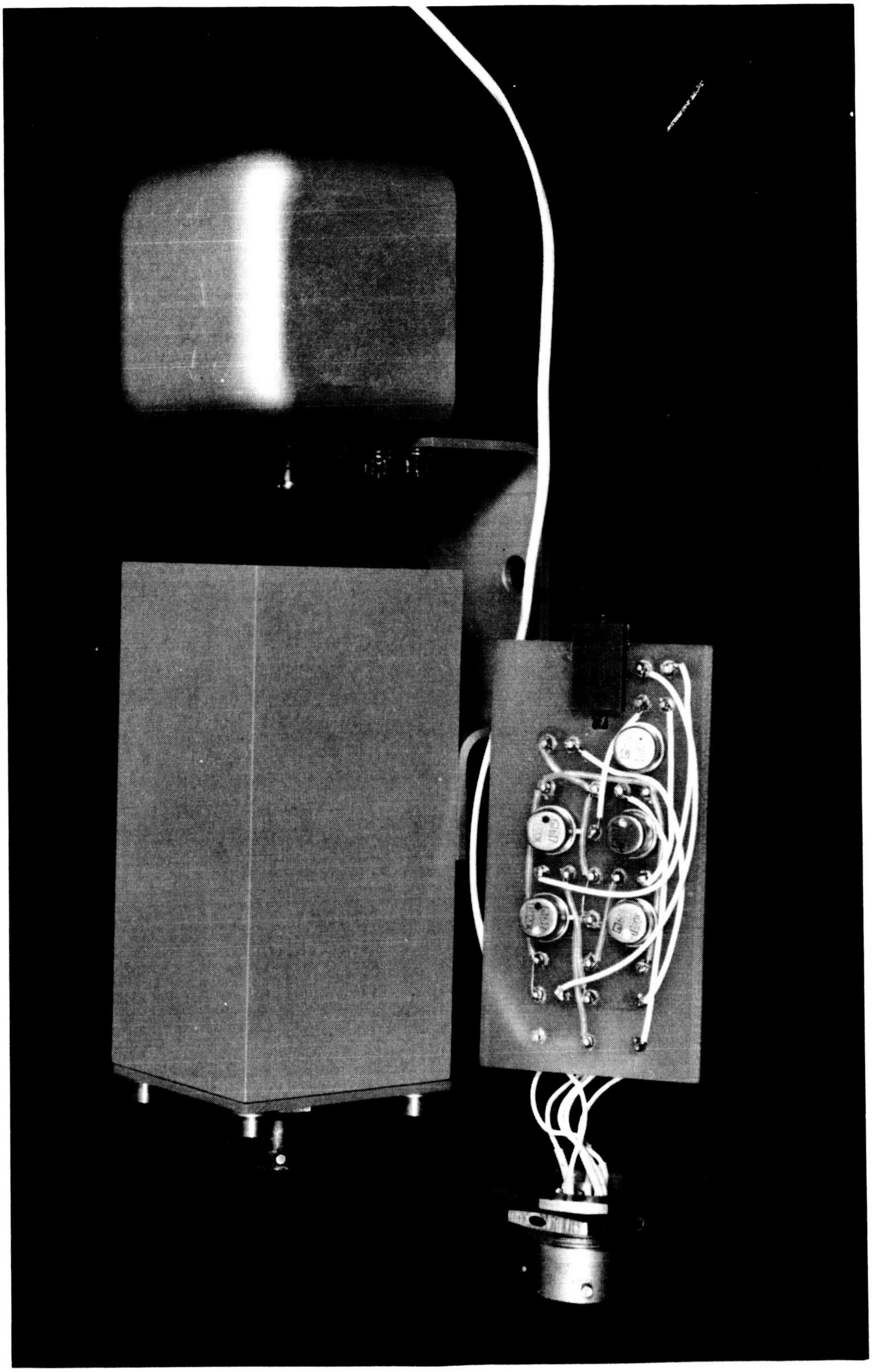


Fig. 16 Pulser Unit Showing Circuit Board and Panel Light

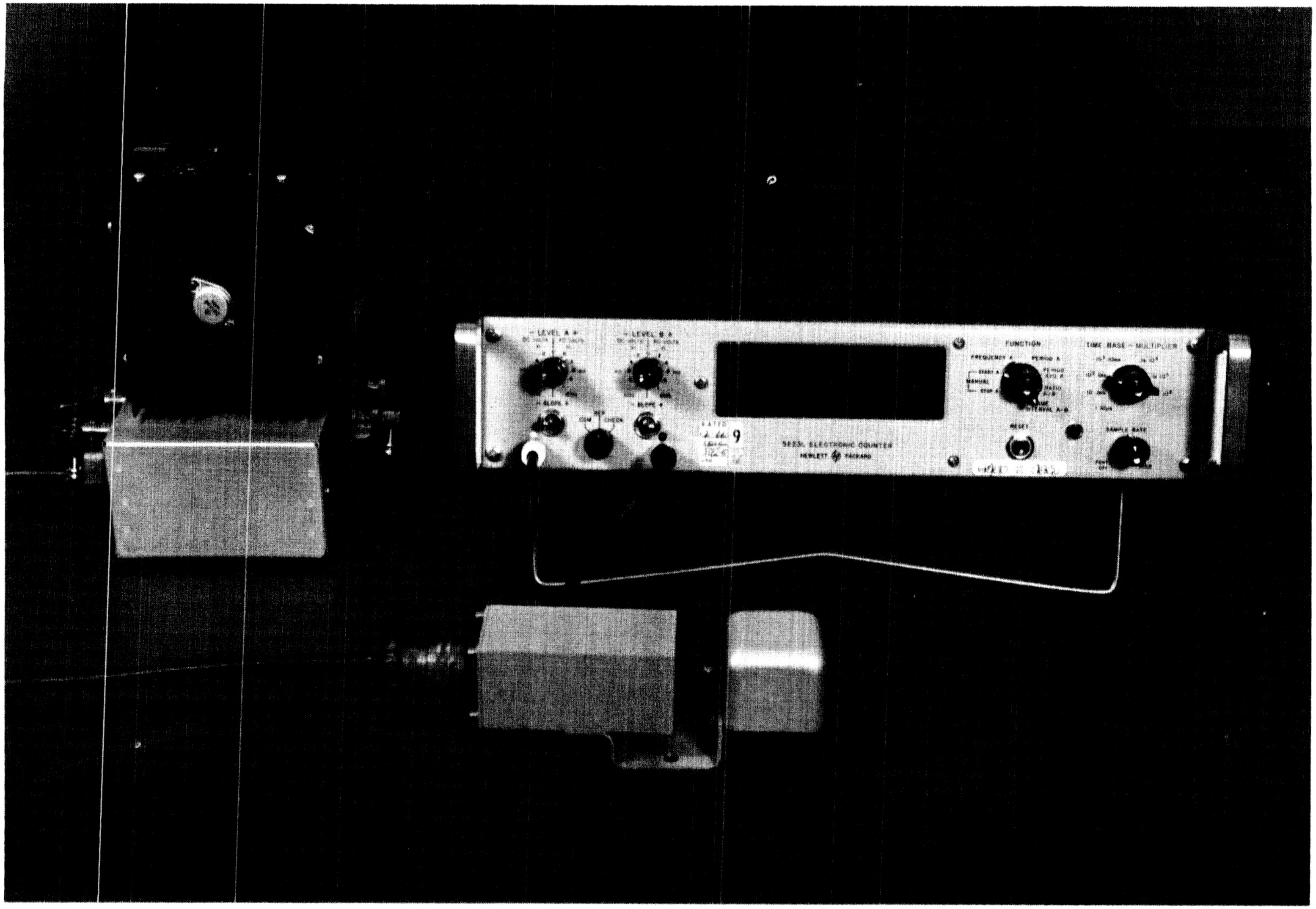


Fig. 17 Electronic Timing System

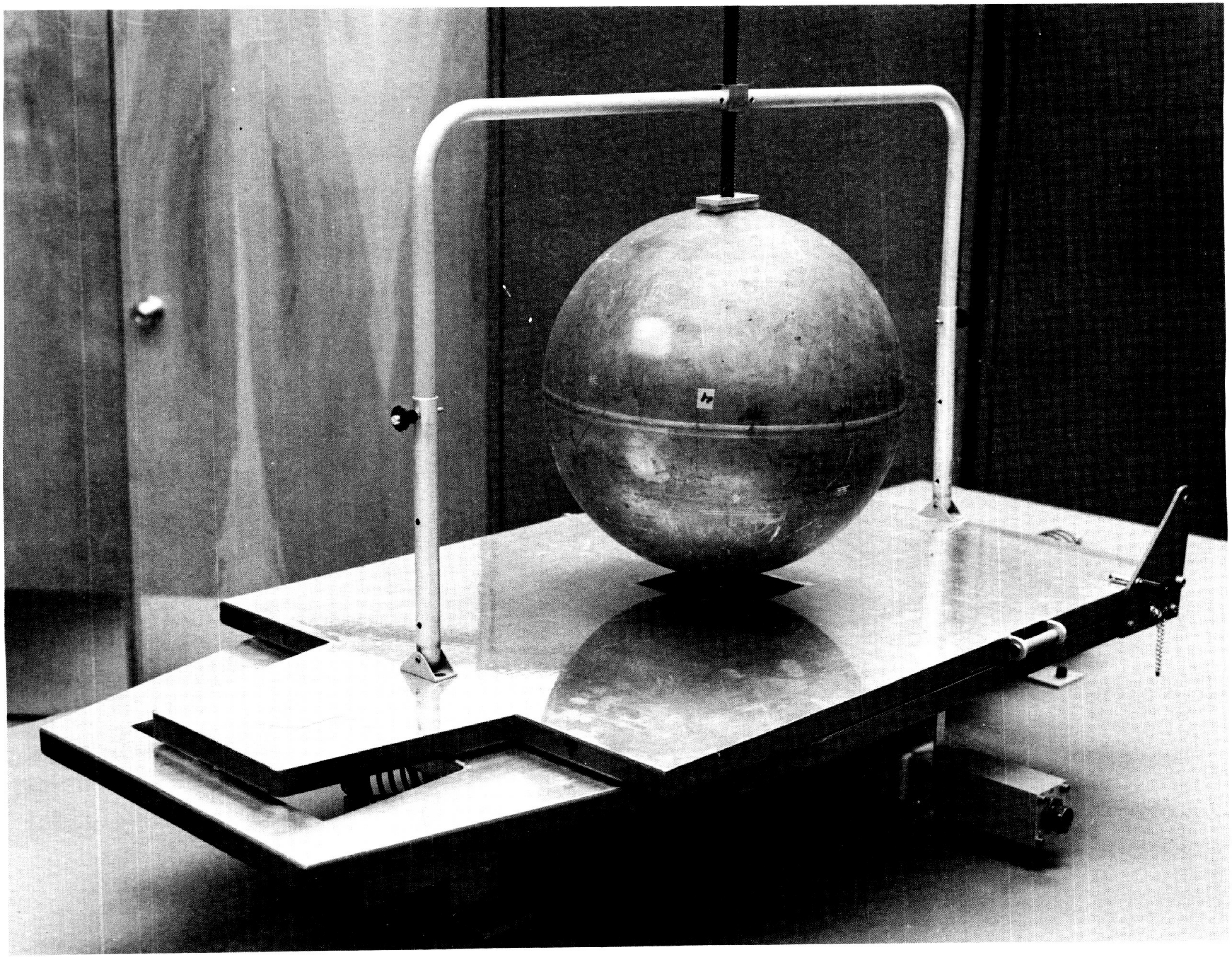


Fig. 18 Inanimate Object Restraint System

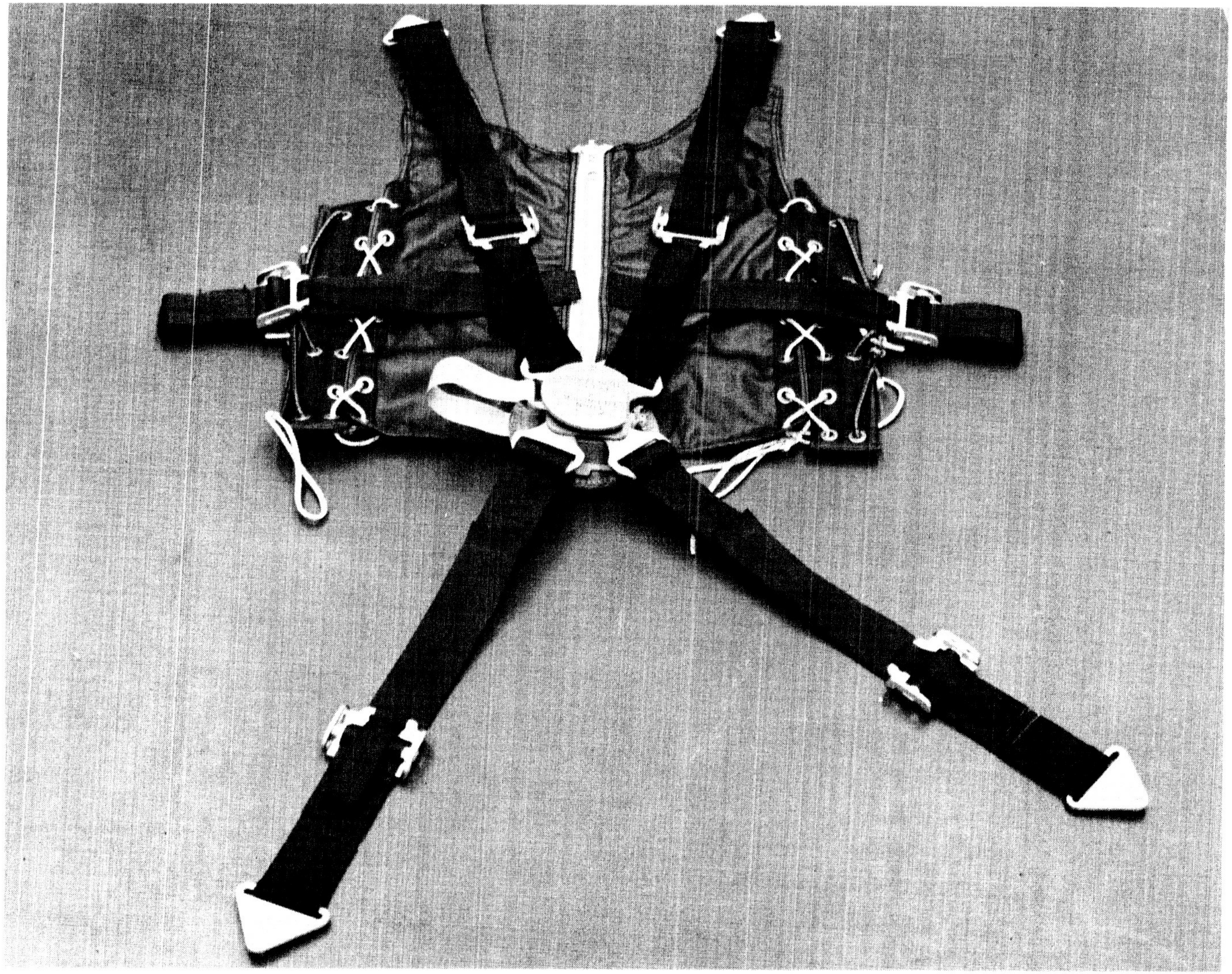


Fig. 19 Man Restraint System

TEST PROGRAM

The primary purpose of the test was to demonstrate the capability of the MMS to make mass determinations of inanimate objects and human subjects. The test program was based on the most realistic requirements envisioned for the MMS for spaceflight utilization.

Test Plan

A detailed test plan was developed for this project. This plan encompassed two major areas: development tests and qualification tests. Development tests consisted of tests of a design-support nature and were carried out concurrently with design. Four investigative categories were covered: (1) material, (2) breadboard, (3) tradeoff studies, and (4) development demonstration.

Qualification tests consisted of functional performance and environmental evaluations which formally demonstrated that the final prototype unit met specification levels and functions in its expected use environment.

Test Procedures

The test procedures prepared for this project were based upon the calibration requirements, preliminary tests, and design verification tests specified in the test plan. Special attention was given to providing standardized test procedures that could be followed during the system and engineering checkout phase planned at the NASA-Langley facility.

Test procedures included:

- Assembly of the MMS in either the -501 (conventional bearing) or the -503 (Saginaw bearing) configuration on a plywood base
- Leveling the assembly in two directions
- Performing detailed mechanical and electrical operational checkouts
- Conducting calibration runs in the 5 to 40 lb range, and in the 41 to 250 lb range (certified weights to be used)
- Checkout of the man restraint harness for fit and function
- Conducting calibration runs in the 100 to 250 lb range using human subjects
- Investigations as to the capability of the system to weigh miscellaneous spacecraft parts, including partially filled containers of liquid

- Determination of effect of eccentric pallet loading and the effect of lateral and longitudinal tilt of the unit

- Design verification tests such as the determination of the mechanical cocking force, total travel distance and spring rates

- Determination of the optimum number of oscillations to be timed

- Establishment of stowed volume, clearance envelope and length

- Mass verification runs using calibrated data

- Dynamic mass measurement verification run using container filled with water

Test Objectives

The overall test objectives were that the MMS must be capable of: Weighing objects in the 5 to 40 lb range, and in the 41 to 250 lb range, functioning when exposed to eccentric pallet loading and lateral and/or longitudinal tilt similar to that experienced in a KC-135 zero-g flight trajectory, and providing data readout accuracies of ±0.25 lb on a rigid mass of 20 lb and ±0.50 lb on a rigid mass of 100 lb. Other test objectives were concerned with establishing the effect of the restraining harness on mass determination, and examining the capability of the system to weigh miscellaneous masses such as partially filled containers of liquid. The design verification tests were oriented toward establishing the feasibility and practicality of adapting the oscillating spring-mass technique to a flight system.

This section details the tests conducted and the results. The detailed test plan established earlier in the project was followed and results obtained met or exceeded performance specification.

Tests

The following major tests were conducted: calibration runs (rigid mass), calibration runs (human subjected), miscellaneous spacecraft parts, including partially filled containers of liquid, tilt and off-center load tests, design verification, optimum oscillation, stowed volume, clearance envelope and length verification, mass verification runs, and dynamic mass measurement verification.

Calibration runs (rigid mass). – Calibration was performed by installing certified weights obtained from the Measurement Standards Laboratory on the MMS. The device was cocked and released and the oscillation time recorded. In the 5 to 40 lb range, weights were increased in 5 lb increments for a total of eight different points. In the range from 41 to 250 lb, weights were increased in 25 lb increments for a total of nine different points. Six runs at each weight were conducted. These runs were done on both the -501 and -503 assemblies. Because of the accuracy requirement it was decided to use a computer program with printout every 0.25 lb for the lower range and 0.50 lb for the upper weight range.

Calibration runs (human subjects). — Six subjects were selected to determine the capability of the MMS to accurately weigh human beings. Weight range between 130 lb to 210 lb was examined since that approximates the 5th to 95th crew percentile. Subjects were weighed on a physician's office scale. The scale was tested for accuracy by using certified weights. The identical procedure used in the calibration runs for the rigid masses was followed for the human subject runs. Six runs for each subject were conducted with runs on both MMS configurations. The same computer program used in the rigid mass calibration was used for the human subject runs.

Miscellaneous spacecraft parts. — A pressure vessel was selected as a typical spacecraft part to examine the capability of the MMS to determine its mass. The vessel was attached to the back of the pallet using the regular tie-down attachment provided for that purpose. Six runs were made using both MMS configurations. Next, the vessel was partially filled with water and a stopper placed in the end of the container. The weight of the vessel increased from 10.35 lb to 19 lb. Six runs were made in that particular configuration. The damping action of the liquid which was sloshing out of phase with the frequency of the oscillating platform was noted. The tendency to reduce the period was significant.

Eccentric pallet loading. — To ascertain the effects of tilting and eccentric loads upon performance. Accordingly, both configurations were tilted to pitch angles of 3-1/2 degrees from the horizontal, with the aft end raised, and without any load on the pallet. There was no appreciable effect on the period. The -501 configuration was also operated with a human subject at a pitch angle of 1 degree, with no appreciable effect. In order to prevent any errors which might have been caused by the shutter being located away from the center of oscillation, the shutter was relocated at the new neutral position. Next, the pitch angle was lowered to zero, the shutter repositioned, and each MMS was tilted to a roll angle of 5 degrees from the horizontal, with its left side raised. A human subject was seated on the pallet, and the test results showed considerable difference from those obtained when the devices were leveled.

Both MMS configurations were returned to level position, and a human subject was tested while sitting directly on center, and then several inches off center toward the right. Appreciable discrepancies resulted (particularly on the -501 assembly) apparently due to the twisting moment applied to the bearings.

In a zero-g environment, there will be no such effects due to eccentric gravitational loads, but in laboratory tests of each MMS it is important to avoid lateral tilt of the apparatus and to place all subjects with their cg on the longitudinal centerline. This centering will normally be accomplished for human subjects by use of the restraint harness and for deadweights by the central location of the clamping device.

Design verification. — Preliminary tests were conducted to establish design verification of such items as mechanical cocking force, total travel distance, and spring rates. Calibration techniques used employed a spring scale for the cocking force and a total of 51 lb was determined. Total travel distance was measured and found to be 8-3/4 in. both ways. Calibrated weights were employed in determining the spring constants, which were found to be 2.85 lb/in. for each spring.

Optimum number of oscillations to be timed. — The electronic timing system employs a bistable multivibrator ripple counter gated to count a discrete number of pallet cycles. By adding an external circuit it was possible to select cycles ranging from 1 to 7 cycles. This range was then examined for the optimum number of oscillations to be timed. In order to simplify the setup for both the -501 and -503 configurations, and afford common ground for comparison purposes, it was decided that the number of cycles selected for one configuration would be used on the other configuration. The physical limitation on the maximum number of oscillations is determined by the configuration having the most friction, aerodynamic drag, and the maximum mass to be determined.

Configuration -501 was selected as the critical system. A load of 250 lb was placed on the pallet in the backup position. Cycles ranging from 1 to 7 were examined. All cycles were counted; however, a marginal condition existed in the upper range. Reproducibility appeared excellent in all conditions. An advantage appears in multiple cycles which affords the opportunity to average out errors. Three cycles were selected as the optimum number and all calibration runs were made at that setting.

Stowed volume, clearance envelope, and length. — The design goals of the MMS included the requirement that the system be stowed in a volume not to exceed 3.0 cu ft. It was also required to be able to pass the MMS in the dismantled condition through a 32 in. diameter circular port.

The MMS in its operating condition has the following basic dimensions:

Back Down – Uncocked	49" x 11" x 20"	Volume 6.2 cu ft
Back Down – Cocked	44" x 11" x 20"	Volume 5.6 cu ft

A considerable reduction in volume is obtained by packaging and stowing the MMS in two separate sections, the pallet and the carriage. This resulted in the following basic dimensions:

Back Down – Pallet only	5" x 20" x 37"	Volume 2.1 cu ft
Carriage	6-1/2" x 8" x 43". . . .	Volume 1.9 cu ft
Total volume		4.0 cu ft

Of the above volume 1.6 cu ft is unoccupied. This unoccupied volume occurs because of the corner gussets that extend approximately 3 in. above the pallet and the carriage plate that is approximately 8 in. diameter. It is practical to provide quick disconnect fittings that would permit stowing these items in much smaller volumes. With this consideration the stowed volume is reduced to 2.4 cu ft which is well below the design requirement.

The MMS can easily be passed through a 32 in. diameter circular port as two principal dimensions are 20" x 11". The length of the MMS (49 in.) is well within the contract requirement of 6.5 ft.

Mass verification runs. — A calibrated rigid mass of 20 lb was weighed to demonstrate the accuracy of the system. A calibrated rigid mass of 100 lb was also weighed. Both tests demonstrated that the MMS (both -501 and -503) was capable of providing accurate mass determination well within the design goal.

Dynamic mass measurement. — A container filled with water weighing approximately 20 lb was weighed on the MMS to determine if the accuracy of the system is within the design goal of ±0.5 lb and a human subject weighing approximately 105 lb was weighed to determine if the accuracy of the system is within ±1.0 lb. Both tests demonstrated that the MMS met specifications.

Calibration Data Analysis

Rigid mass calibration. — Models -501 and -503 were calibrated for rigid-body weights using a set of known (1 part in 10,000 accuracy) weights. The electronic timer used reads to ±0.1 millisecond accuracy. The minimum period of interest is about 2,000 msec, yielding a minimum timing accuracy of 1/2 part in 10,000. The low and high weight ranges were separately calibrated. Results are given in Table 6 in which the mean time interval for the chosen number of oscillation cycles is given, along with the range. Each point was obtained six times. Reproducibility is excellent as shown by the maximum standard deviation of the sample data for Model -501, of 2.8 parts in 10,000. This indicates that there is a 99 percent probability that any single time obtained is ±7.22 parts in 10,000 within the same time. Corresponding values for Model -503 are maximum standard deviation of 5.4 parts in 10,000, and 99 percent probability that a time is within ±13.9 parts in 10,000.

As the weighing accuracy desired is a minimum of 0.5 lb in 250 (or 20 parts in 10,000) and as sample statistics indicate an order of magnitude of better reproducibility, interpolation using a digital computer program was judged worthwhile. First the calibration points are fit (using a least squares routine) to a polynomial expression (up to fourth order polynomials).

The program matches polynomials in x, exp x, ln x, 1/x, 1/exp x, 1/ln x versus an analogous set of six expressions in y, ranks the best 10 of the 144 cross matches, and calculates coefficients and deviations from the calibration points. The best match is then used in a second computer run and the interpolation tables prepared.

These interpolation tables were tested using a series of initially unknown weights, in the range of 1/4 to 50 lb. Weight to the nearest 1/4 lb can be immediately determined by inspection of the interpolation table. Using linear interpolation between tabulated values, weight to the nearest 0.05 lb can be calculated quickly. The beam balance used to check these weights was not accurate enough to verify the interpolated values, but the MMS weight value agreed with the weight value read from the balance within the 0.10 lb to which the beam balance could be read.

Human subject calibration. — Calibration data for human subjects was obtained in a manner analogous to that used for rigid-body calibration. Subjects are weighed immediately before taking the MMS data, using a beam balance previously standardized and found accurate to the 0.1 lb to which it could be read. All subjects wore the restraint harness. Six replicates of period for 3 cycles of oscillation were taken. Table 6 gives the mean time, and the range, which is significantly larger than for rigid masses. Standard deviation is estimated using the approximation σ = range/2.534. Maximum standard deviation for model 501 is 6.5 parts in 10,000, or a 99 percent probability

Table 6

MMS CALIBRATION DATA – RIGID MASS

Model -501 – 3 Cycles

Range 5 – 40 lb Back Down			Range 40 – 250 lb Back Up		
Weight (lb)	Period (sec)	Range of Period (6 replicates-msec)	Weight (lb)	Period (sec)	Range of Period (6 replicates-msec)
0	1.86643	0.6	0	1.88928	0.6
5	2.07022	1.0	30	2.89293	1.3
10	2.25647	1.3	50	3.40352	0.9
15	2.43033	1.7	75	3.94495	2.6
20	2.58818	1.0	100	4.42220	0.9
25	2.74260	2.1	125	4.84933	0.9
30	2.88418	1.8	150	5.23812	2.3
35	3.01922	0.7	175	5.60637	3.9
40	3.14988	1.8	200	5.94323	2.9
45	3.27600	2.0	225	6.26513	4.3
50	3.39295	1.9	250	6.56350	3.2

Model -503 – 3 Cycles

Range 5 – 50 lb Back Down			Range 40 – 250 lb Back Up		
Weight (lb)	Period (sec)	Range of Period (6 replicates-msec)	Weight (lb)	Period (sec)	Range of Period (6 replicates-msec)
0	1.88263	2.6	0	1.91696	2.7
1	1.92148	1.7	1	1.95728	2.1
5	2.07962	1.5	30	2.89217	0.9
10	2.26157	0.8	50	3.38818	5.0
15	2.42838	1.9	75	3.92117	2.2
20	2.58465	0.7	100	4.38522	5.8
25	2.73407	3.2	125	4.80507	4.8
30	2.87268	1.7	150	5.19253	8.0
35	3.00830	2.0	175	5.54690	6.5
40	3.13538	0.8	200	5.87902	4.0
45	3.25760	1.9	225	6.19632	2.1
50	3.37520	2.1	250	6.49845	6.4
			251	6.50757	8.3

that a single period measurement is within 16.8 parts in 10,000 of the mean. For model -503, standard deviation is 8.2 parts in 10,000 or a 99 percent probability of 21.2 parts in 10,000 deviation.

The desired weighing accuracy is ±1 lb in 250 lb or 40 parts in 10,000. Statistics relative to this goal are again favorable, and a digital computer data fit and interpolation table calculation is used. Table 7 gives the results for a subject weight range of 100 to 250 lb. Weights beyond the 5th and 95th percentile are extrapolated. Accuracy in these ranges may be lower than within the 130 to 200 lb range for which calibration points were taken. Note that the restraint harness weight is included as part of the MMS tare for these human subject tables. If a lighter restraint harness is used, subject weight must be found by adding the weight difference to the calculated weight.

Test Results Summary

Following the functional checkout and the computer calibration analysis several tests were made to demonstrate the versatility of the system and also to establish limitations that may exist. These tests are described under the previous test section. This section will summarize the test results. There were two classifications of test: preliminary and verification. The preliminary tests examined: man restraint system, miscellaneous spacecraft parts, eccentric pallet loading, design verification tests, and optimum number of oscillations. The verification tests covered: stowed volume, clearance, length rigid mass measurements, and subject tests.

Man restraint system. – Runs were made with and without the man restraint system donned. No attempt was made to determine quantitatively the difference the restraint system made on mass determination. It was obvious after several measurements with the restraint system off, that the period was markedly affected by the subject's attitude and rigidity. It is recommended that the restraint system be donned during operational runs.

Miscellaneous spacecraft parts. – A pressure vessel was used to demonstrate the capability of the system to weigh typical spacecraft parts. The pressure vessel weighed 10.35 lb. The second part of the test was to partially fill the vessel with water which increased its weight to 19 lb. Six runs were made in both configurations. The range of period for six replicates was 2.0 msec which was in good agreement with the rigid mass calibration runs. Using the average period and entering the calibration tables it was determined that the weighing showed a 10.6 lb for the empty pressure vessel both on the -501 and the -503 assembly. This weight is within the specified tolerance of ±0.25 lb.

The partially filled liquid container runs showed a much greater range of period for six replicates (13.0 msec). This was to be expected because of the sloshing effect. Entering the calibration tables, the period measured indicated that the partially filled container weighed 24.1 lb. This exceeded the actual weight by 5.1 lb. Random effects due to sloshing of the liquid cause unpredictable results. This would imply that it would be most difficult to weigh a liquid filled container without detailed information on the container, volume occupied, density of the liquid, etc.

Table 7

MMS CALIBRATION DATA – HUMAN SUBJECTS

Model -501 – 3 Cycles Back Up

Weight (lb)	Period* (sec)	Range of Period (6 replicates-msec)
130.4	5.03710	8.0
144.5	5.25887	2.1
159.75	5.47670	2.4
174.0	5.68075	2.4
184.4	5.83095	6.9
195.4	5.97952	5.7

Model -503 – 3 Cycles Back Up

Weight (lb)	Period (sec)	Range of Period (6 replicates-msec)
108.2	4.64742	4.6
130.4	5.01232	10.4
143.2	5.21078	9.0
158.9	5.44537	9.6
171.75	5.62060	6.1
184.25	5.79493	9.0
193.6	5.91572	9.9
201.2	6.00368	9.4

*Last digit not significant.

Eccentric pallet loading. – Tests were conducted with and without a subject. Six replicates in each configuration and in each orientation were made. No appreciable effect on the period was experienced with the MMS tilted to pitch angles of 3-1/2 degrees (without load) and one degree (with human subject) on the pallet. Test results showed considerable differences when the MMS was tilted to a roll angle of 5 degrees and a human subject seated on the pallet.

A subject was placed off center toward the right with both MMS in the level position. Results were appreciably different from those obtained with the subject centered. This was apparently caused by the twisting moment applied to the bearings. Test results have led to the recommendation that in laboratory tests of each MMS it is important to avoid lateral tilt of the apparatus and subjects should be placed on the longitudinal centerline.

Design verification. – Measurements were made for mechanical cocking force, travel distance, and spring rates. Data is tabulated in the design specification table.

Optimum number of oscillations to be timed. – Test results indicated that 3 cycles would be optimum for both MMS configurations. The electronic pulser has the capability of counting any number of cycles from 1 to 7.

Stowed volume, clearance envelope, and length. – Measurements were made of the stowed volume, clearance envelope, and length. Data is tabulated in the design specification table.

Rigid mass measurement. – A rigid mass of 20.0 lb was weighed on both configurations and the calibration table used to determine the weight. Six replicates showed a range of period of 1.0 msec which was consistent with data compiled in the calibration runs. The calibration table yielded a weight of 20.25 lb, which is within the ±0.25 lb requirement.

A rigid mass of 100 lb was weighed on both configurations and the calibration table used to determine the weight. Six replicates showed a range of 1.0 msec which was consistent with data compiled in the calibration runs. The calibration table yielded a weight of 100.40 lb, which is within the ±0.50 lb requirement.

Dynamic mass measurement. – Two plastic containers filled with water and weighing 20.25 lb were weighed on both configurations as part of the design verification. On the -501 assembly, six replicates showed a range of period of 0.7 msec which was well within the calibration data compiled. The calibration table yielded a weight of 20.6 lb or +0.35 lb, which is slightly above the ±0.25 lb requirement.

On the -503 assembly, six replicates showed a range of period of 3.7 msec, which is above the calibration data compiled. The calibration table yielded a weight of 21.1 lb or +0.85, which is above the ±0.25 lb requirement.

This test indicated that liquid mass determination may present problems as far as meeting accuracy requirements of ±0.25 lb. This is probably caused by the plastic containers deforming slightly under acceleration. Special attention was given in this test to remove any air space; however, any sloshing would directly affect the period reading.

A human subject weighing 108.2 lb was weighed on both configurations. Six replicates showed a period range of 2.5 msec which is in good agreement with the calibration data. The calibration table yielded a weight of 109.0 lb which is within the ±1.0 lb requirement.

Design specifications. — The overall concept as well as the design details of this MMS were developed to meet the requirements and the design goals needed by NASA, as outlined in the original Work Statement for this contract. The requirements have been met, and the design goals were exceeded in some instances, but were not quite achieved in others.

It is believed that this MMS concept is suitable for spacecraft equipment. Table 8 is a summary of the specifications of this MMS. -501 designates the conventional Fafnir ball bearing carriage, and -503 designates the Saginaw linear ball bearing carriage configuration. Undesignated specifications apply to both.

Table 8

DESIGN SPECIFICATIONS

	Configuration -501	Configuration -503
Will pass through 32 in. dia port, required	yes	
Length: 78 in. max required – stationary	48 in.	49-1/2 in.
operating	56-3/4 in.	58-1/4 in.
Volume: 3 cu ft storage. Goal	*	
Weight: 25 lb goal (no pwr supply, spacer, counter or harness)	33.6 lb	38.6 lb
Operating temperature range 70 to 120°F. Goal	yes	
Electrical power needed – for pulser	28v dc	
for timer readout	115v dc 60 cycle	
Portable. Goal	yes	
Simplicity of operation. Goal	No changing of springs required over complete range. Capable of one man operation.	
Human comfort. Goal	yes	
Gravity environment. Zero to 1 g, operating Goal.	Calibration applies at 1 g, vertical. Would change for zero g or lateral tilt.	
Flight loads of 5 g. Goal	Requires support blocking.	
Range, 5 to 250 lb. Goal (Conversion permissible)	1/4 to 250 lb (No conversion needed)	
Accuracy. Rigid mass. 40 lb. Goal ±1/4 lb	±1/4 lb	
41 to 250 lb. Goal ±1/2 lb	±1/2 lb	
Accuracy. Human subject. Goal ±1 lb	±1 lb	
Cocking force	51 lbs	
Cocking distance	8-3/4 in.	
Spring constant (of each spring)	2.85 lb/in.	

*Depends upon criterion used. See page 58.

PRELIMINARY DESIGN KC-135 FLIGHT PROTOTYPE MMS

The purpose of the KC-135 flight prototype MMS is to check out the operational performance of the oscillating spring mass system technique in the zero gravity environment produced during KC-135 aircraft flight parabolas. Further investigation for test subjects will include physiological effects resulting from the linear and oscillatory motions.

This section of the report points out the major design differences between the prototype MMS and the KC-135 flight version. The major design consideration which affects the hardware is the added g loads imposed during the flight parabola. These loads can reach between 3 and 3.5 g during pullout before and after the parabola. The MMS work statement specifies that the MMS be designed structurally for any condition of gravity from zero to 5 gs. If a 250 lb man is seated on the pallet and a 1.5 margin of safety added in conjunction with the 5 g load, the total load on the MMS becomes 1,875 lb. To withstand this load the pallet has to be redesigned. The existing carriage design is adequate if a block is placed between the lower surface of the rail and the floor of the KC-135 aircraft. This is to reduce the high bending loads on the rail preventing deflections which might cause inaccuracies in the oscillating system and reduce total system weight. It is interesting to note, that while the loads imposed on the MMS are high just before and after a zero-g KC-135 flight parabola, the loads imposed on an actual spacecraft flight MMS would be those of booster launch, during which time the MMS can be stowed and locked in place. Since the operation of MMS in a spacecraft or space station on orbit would be exposed to loads ranging from zero to perhaps 1 g the structural requirements are much less. This fact will allow the system to be much lighter and can be designed within the 25 lb weight goal.

Pallet. – The main change in the pallet design is the increased depth of the honeycomb structure. As can be seen in fig. 20, the structure is 2.00 in. thick at the carriage attach point. This tapers to 1.00 in. thick at the back and foot. In addition, the back structure has been increased from .75 to 1.00 in. Channels have been added to the aft edge of the pallet seat and also to the lower edge of the pallet back. The skin thickness is .025. The bonding process is unchanged.

Restraint system. – No changes in the restraint system for both animate and inanimate objects is planned. The man restraint system remains unchanged. However, if future testing with the prototype MMS indicates changes are required in this restraint system for specific items of interest, improvement could be incorporated in the new design as required.

Carriage. – A considerable amount of weight can be saved if various component parts will not be subjected to a 5-g load with a man seated on the MMS. This can be rather simply accomplished by providing removable supports for the pallet seat (or the bearing carriage). These supports will transfer the seat loads directly to the floor of the aircraft, rather than through the bearings and rail.

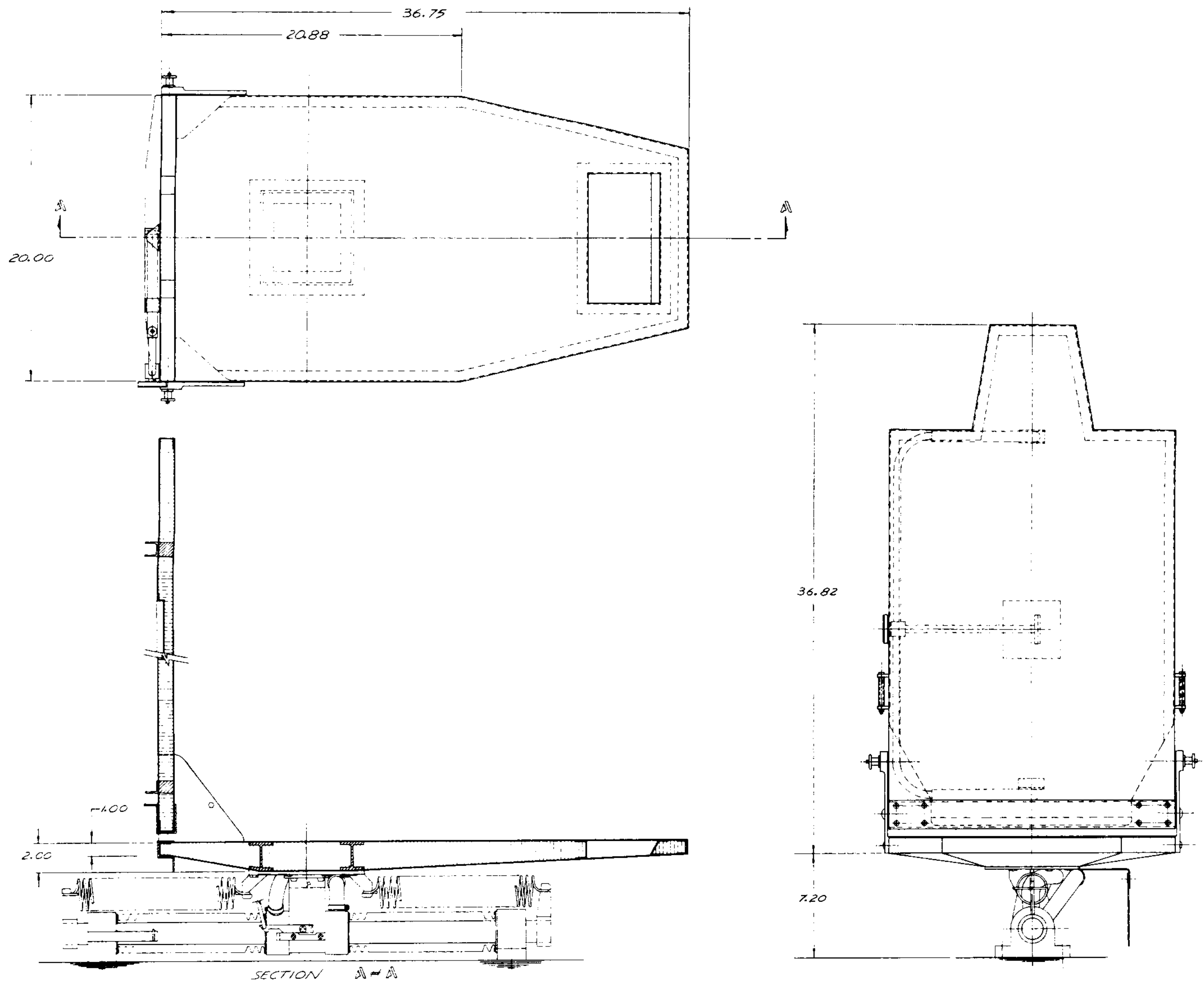


Fig. 20 Preliminary Design of the KC-135 Prototype MMS

Both height and length can be reduced by locating the springs off center; one to the right and one to the left of the carriage. With this arrangement the height of the springs and the rail will not be additive since they will not be above one another. Similarly, the length of the two springs will not be additive since they will overlap one another. The twisting movement applied to the bearing will be consistent and should therefore not diminish accuracy.

Electronics improvements. — Improvements in the MMS electronics portion relate to using improved packaging techniques and reducing power consumption.

The use of printed circuitry would permit a smaller electronics board assembly. This would also allow for the fabrication of a smaller and lighter housing. Fabrication of the housing could be simplified by making it of half sections which could then be fastened together with the board assembly. Excess material around the housing would be removed, thereby further reducing weight.

The power consumed by the electronics package is mainly used by the light source to activate the phototransistor. This power (approximately 3 watts) could be reduced at least an order of magnitude by the use of a focusing lens. Such a lens could be part of the bulb, such as General Electric's #222. In addition, a lower power light source could be placed in a small package, because of reduced thermal dissipation.

Stiffening the mounting bracket is recommended. Errors in operation of the electronics due to a wandering light path would be precluded since the bracket aligns the light source with the phototransistor. Adding two side members should be sufficient.

Structural Analysis

A preliminary analysis to determine the material sizes for the MMS has been completed. The structure was analyzed for the flight loads realized when a KC-135 aircraft is flown through a zero-g flight parabola. The assumed loads and recommendations for a structure capable of sustaining these loads are contained in the discussion below.

Loads. — The following assumptions on loads were used to analyze the mass measurement system:

- The weight of a person on the pallet will not exceed 250 lb.
- 0.7 of the weight can simultaneously act on the back and bottom of the pallet. This will approximate a weight vector at 45° to the bottom of the pallet.
- The limit dynamic load factor is 5.0 g's.
- Factor to ultimate is 1.5 times limit.

Analysis. – The purpose of the analysis was to determine the necessary structure for sustaining the loads previously quoted.

The materials for the structure are Military Grade 5052 alloy aluminum honeycomb, designation 1/4-5052-.002 with 6061-T4 facing material. All material properties utilized are minimum properties as per Hexcel Products TSB 120. The loads applied normal to the back of the pallet must be taken out in a uniform manner along the bottom edge. Otherwise, local failure will take place at the corners containing the gusset connections. A structural assemblage capable of doing this is shown in fig. 21. Also shown in fig. 21 is the local reinforcement for the gusset connection and the actual cross section dimensions for the back of the pallet. The connectors from the back to the gusset must be capable of withstanding an ultimate shear load of 3750 lb each.

The transfer of the "back" loads to the bottom of the pallet must also be in a uniform manner. A concept similar to that shown in fig. 21 for the back, would be suitable for the bottom.

The seat attaches to and rides on a rail. During maximum g, the rail deflections become excessive. Hence it is essential that the rail be blocked during peak loads.

Conclusion. – It is quite evident that the prototype mass measurement system is structurally inadequate for the KC-135 flight parabola. The structural modifications suggested will be the minimum necessary to make it structurally adequate.

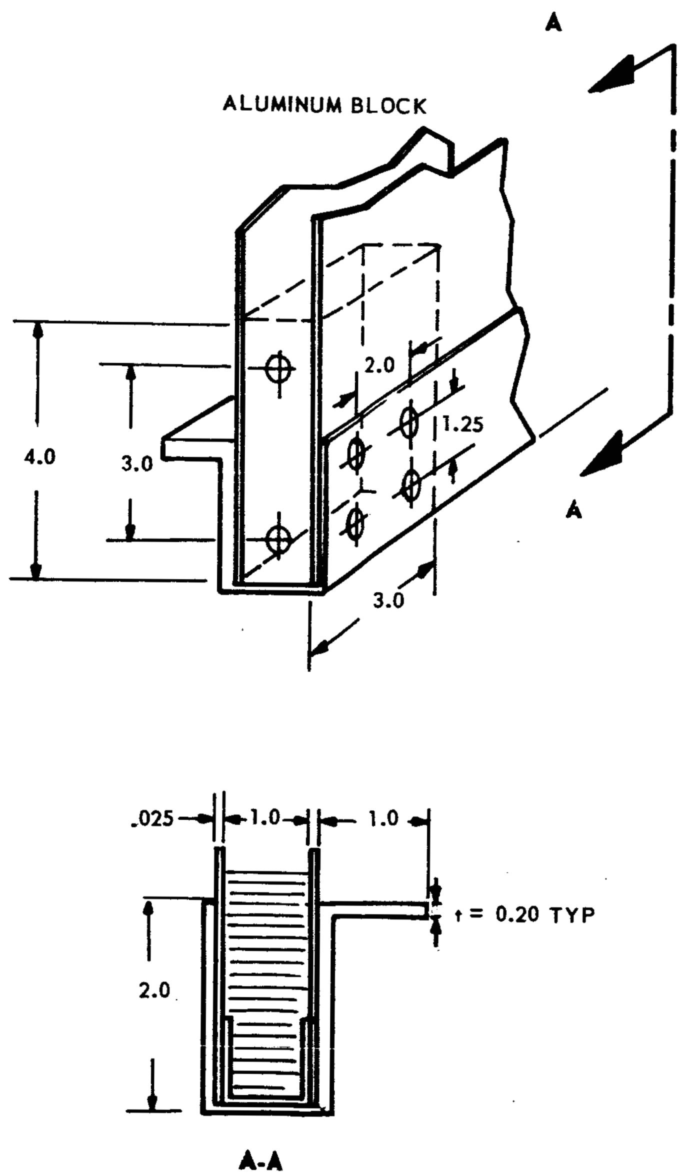


Fig. 21 Structural Assemblage for Pallet Back

CONCLUSIONS AND RECOMMENDATIONS

The MMS program has demonstrated that a device can be designed and fabricated to determine mass, utilizing the oscillating spring mass technique within the accuracy goals required. Both carriage systems used work within the requirement envelope under the conditions of the 1 g environment. The Model -501 carriage system operates more smoothly than the Model -503 carriage system and with less noise, an important point, when the design is considered for use in measuring acceleration thresholds of human subjects. The Model -501, statistically, shows a slightly higher probability of repeating the same readout for a given mass than does the Model -503. The Model -501 does have more friction which causes it to dampen faster. It also does not work as well when a torsional load is applied to the carriage.

The Model -503 carriage system is not as sensitive to torsional loads as the -501 system. The -503 system produces considerable noise during oscillation but has much less friction. To select the best system, further testing in the zero-g environment would provide more data points on which engineering analysis could be based. The KC-135 flight program would be required for this evaluation.

The use of the honeycomb structure proved to be an adequate material from which to fabricate the pallet. However, it is recommended that thicker skin panels be used, even at the cost of additional weight to withstand the rough usage of placing heavy and sharp-edged objects on the pallet. The present skin thickness is .012. The use of .025 skin panels is recommended.

The restraint systems for both man and inanimate object is considered adequate. Design improvement could be optimized, if actual items of interest could be defined prior to a given space mission and the restraint system modified for the given tasks.

The electronic timing system proved reliable and future designs should be based on the existing system. It is recommended however, that the flight system readout be in direct pounds converted by an on-board spacecraft computer system. Studies should be initiated on the integration of such a system.

The forthcoming interim period of design evaluation by the NASA-Langley engineering team presents an opportunity for the LMSC design engineering team to develop solutions for problems known to exist for the space flight version. The following tasks would be investigated to insure a timely systematic approach to end-item spacecraft hardware:

- Fabricate the KC-135 zero-g flight version of the MMS utilizing the existing carriage systems with a new pallet assembly as outlined in the preliminary design section of this report
- Perform KC-135 flight test program and design evaluation
- Spacecraft experiment definition
- Experiment equipment/spacecraft interface analysis

- Threshold acceleration adaption requirements
- Design analysis
- Preliminary design
- Fabricate functional mockup of MMS for space flight use
- Development plan for flight prototype

These tasks would be integrated to meet the requirements of the Apollo application program where applicable.

With the MMS concept proven within the accuracies required, the next step in the hardware development would be the fabrication of a KC-135 flight prototype verifying performance characteristics while flying a zero-g parabola, and the preliminary design of a spacecraft/space station oriented mass measurement system as pictured in fig. 22.

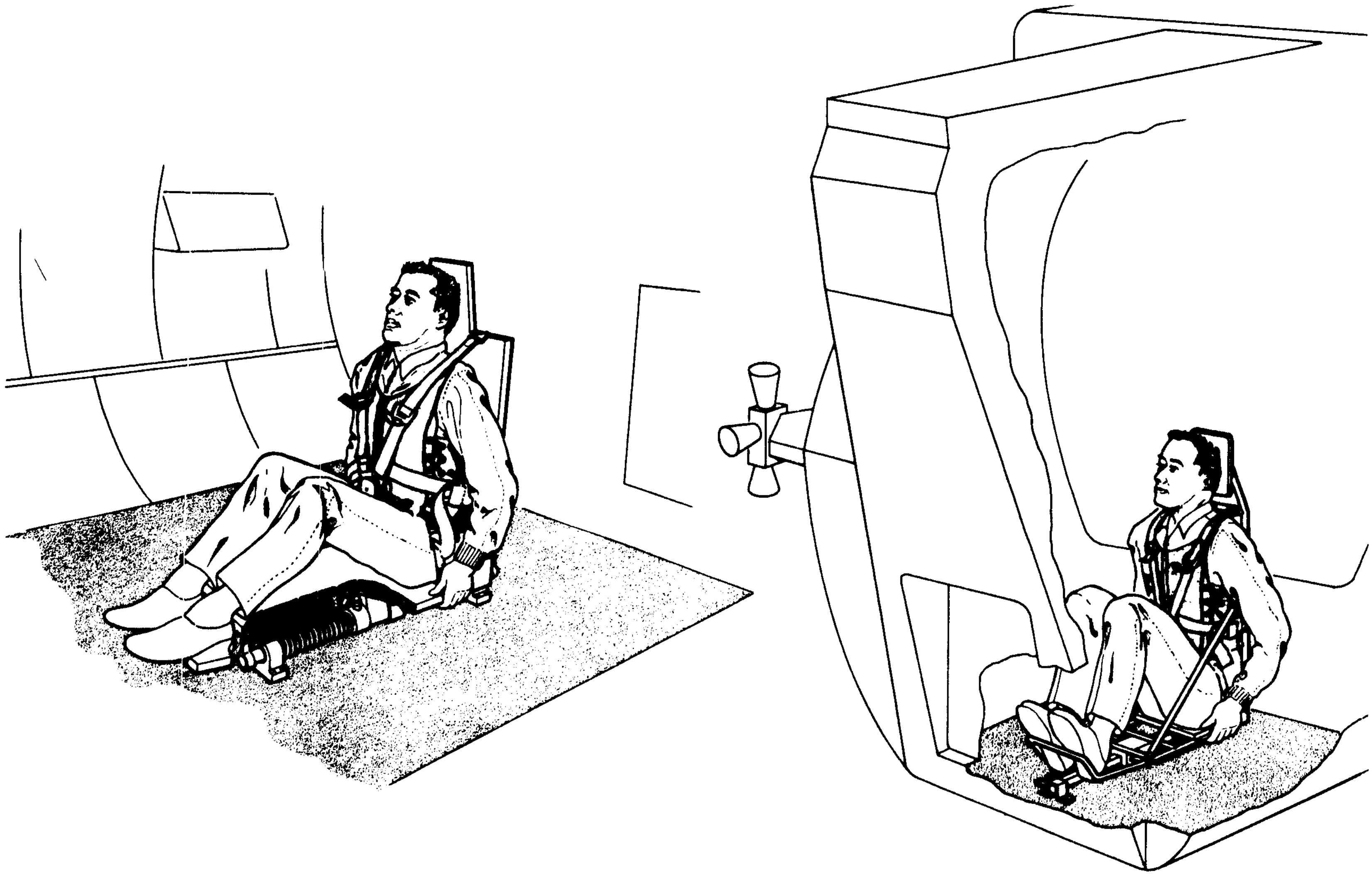


Fig. 22 KC-135/Spacecraft MMS

APPENDIX A

METHODS OF GENERATING LINEAR ACCELERATION

The linear acceleration of a moving body, at any instant, is defined as the time rate of change of the linear velocity of the body at the instant. Since velocity is a vector quantity, the change, ΔV, may be due to a change in the magnitude only of the velocity as in rectilinear motion with varying speed; or to a change in the direction only, as in curvilinear motion with constant speed; or to a change in both magnitude and direction as in curvilinear motion with varying speed. The acceleration of a body having various types of motion will be considered.

Acceleration in rectilinear motion. – A body moves so that its velocity changes in magnitude only. Acceleration can be presented as:

$$a = \frac{\Delta v}{\Delta t} = \frac{v_2 - v_1}{t_2 - t_1} = \frac{dv}{dt} = \frac{d^2 s}{dt^2} = v\frac{dv}{ds}$$

Figure A-1(a) illustrates this type of motion.

Uniformly accelerated rectilinear motion. – Straight-line motion with constant acceleration (i. e., free-falling body) is illustrated in fig. A-1(b). It can be represented by:

$$a = \frac{\Delta v}{\Delta t} = \frac{v - u}{\Delta t}$$

where

$$v = u + at \ , \quad s = \left(\frac{u + v}{2}\right)t \ , \quad s = ut + \frac{1}{2}at^2$$

v = final velocity

u = initial velocity $\quad v^2 = u^2 + 2as$

Acceleration in curvilinear motion. – A point moving with varying speed along a curved path. Linear acceleration's magnitude and direction can be determined by examining the tangential and normal acceleration vectors (a_t and a_n) . Figure A-1(c) illustrates this type of motion.

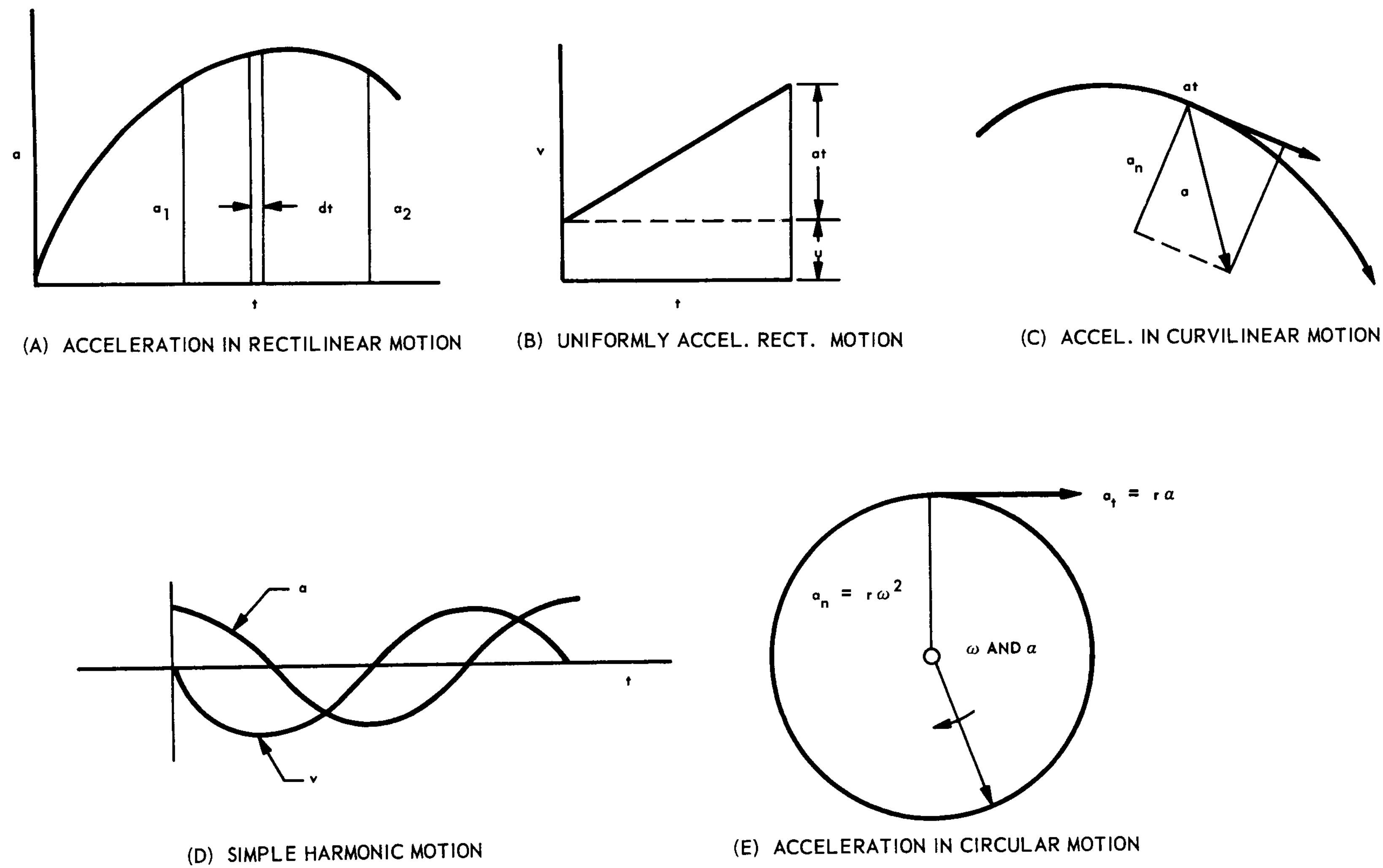


Fig. A-1 Modes of Linear Acceleration

$$a_t = \frac{dv}{dt} = \frac{d^2s}{dt^2}$$

$$a_n = v\omega = \omega^2 r = \frac{V^2}{r}$$

$$a = \sqrt{a_n^2 + a_t^2}$$

ω = angular velocity

Simple harmonic motion. – A special case of rectilinear motion with variable acceleration. The motion of a point in a straight line such that the acceleration of the point is proportional to the distance, x , of the point from some fixed origin, 0 , in the line and is directed toward 0 .

$$a = \frac{d^2s}{dt^2} = -kx$$

k = constant

The motion of an oscillating pendulum also approximates closely a simple harmonic if the arc through which the pendulum swings is small.

If a point moves with constant speed in a circular path, the motion of the projection of the point on a diameter of the circle is a simple harmonic motion. Figure A-1(d) illustrates this type of motion.

$$a = \omega^2 x = \omega^2 r \cos \omega t$$

Acceleration in circular motion. – The motion of a body on a circular path with constant angular acceleration occurs in a centrifuge. The tangential acceleration, at any instant, of a point moving on a circular path is equal to:

$$a_t = r\alpha \, , \quad a_n = r\omega^2$$

where

α = angular acceleration

The normal acceleration, $r\omega^2$ of the body directed along the radius vector is independent of the angular acceleration. It depends on the angular velocity at the instant, and not on the rate at which the angular velocity is changing at the instant. Figure A-1(e) illustrates this type of motion.

Comparison analysis. – A summary of the five types of motion and their characteristics is presented in Table 4. Applying the constraint that the type of motion should be compatible with the MMS as designed would limit the selection to that type of motion based on simple harmonics. However, if components could be removed and replaced on the MMS, such as springs, or a servo motor could be installed, then two other types of motions appear feasible, i. e., acceleration in rectilinear motion, and uniformly accelerated rectilinear motion. The remaining two types of motion, acceleration in curvilinear motion, and acceleration in circular motion can be eliminated from further consideration due to the major modification required to adapt those types of motion to the Mass Measurement System.

REFERENCES

1. Hertzberg, H.: Daniels, G.: and Churchill, E: Anthropometry of Flying Personnel. WADC Tech Report. 52-321, Wright-Patterson AFB, Ohio, September 1954

2. Duddy, J.: Apollo Space Suit and Extravehicular Mobility Unit. LMSC 5-02-66-1, Biotechnology Orgn. Lockheed Missiles & Space Co., Sunnyvale, Calif., March 1966.

3. Hanavan, E.: A Mathematical Model of The Human Body. AMRL TR 64-102, Wright-Patterson AFB, Ohio, October 1964

4. Santschi, W.; DuBois, J.; and Omoto, C.: Moments of Inertia and Centers of Gravity of the Living Human Body. AMRL TR 63-36, Wright-Patterson AFB, May 1963

5. DuBois, J.: Santschi, W.; Walton, D.; Scott, C.; and Mozy, F.: Moments of Inertia and Centers of Gravity of the Living Human Body Encumbered by a Full Pressure Suit. AMRL TR 64-110, Wright-Patterson AFB, Ohio, November 1964

6. Fields, W. S.: Clinical Problems Associated with Prolonged Exposure to Weightlessness. Symposium on the Role of the Vestibular Organs in the Exploration of Space NASA SP-77, pp. 203-207

7. Spoendlin, H.H.: Ultrastructural Studies of the Labyrinth in Squirrel Monkeys. Symposium on the Role of the Vestibular Organs in the Exploration of Space NASA SP-77, pp. 7-22

8. Milojevic, B.: Influence of the Otoliths on the Duration of the Post-Caloric Nystagmus. Symposium on the Role of the Vestibular Organs in the Exploration of Space NASA SP-77, pp. 77-84

9. Johnson, W.H., Money, K.E., and Grabiel, A.: Some Vestibular Responses Pertaining to Space Travel. Symposium on the Role of the Vestibular Organs in the Exploration of Space NASA SP-77, pp. 215-219

10. Guedry, F.E.: Comparison of Vestibular Effects in Several Rotating Environments. Symposium on the Role of the Vestibular Organs in the Exploration of Space NASA SP-77, pp. 243-256

11. Van Egmond, A.A.J.; Groen, J.; and Jongkees, L.B.W.: The Function of the Vestibular Organ. Pract. Otorhinolaryng. (Basel) 14:Supp. 2, 1952

12. Niven, J.I.; Hixon, W.C.; and Correia, M.J.: An Experimental Approach to the Dynamics of the Vestibular Mechanisms. Symposium on the Role of the Vestibular Organs in the Exploration of Space NASA SP-77, pp. 43-56

13. Clark, B.; and Grabiel, A.: Some Factors Contributing to the Delay in the Perception of the Oculogravic Illusion. Symposium on the Role of the Vestibular Organs in the Exploration of Space NASA SP-77, pp. 151-162

14. Miller, E. F.; and Grabiel, A.: Otolith Function as Measured by Ocular Counter-rolling. Symposium on the Role of the Vestibular Organs in the Exploration of Space NASA SP-77, pp. 121-131

15. Grabiel, A; Kennedy, R. S.; Guedry, F. E.; McLeod, M. E.; Colehour, J.K.; Miller, E. F.; Knoblock, E. C.; and Mertz, W.: The Effects of Exposure to a Rotating Environment (10 RPM) on Four Aviators for a Period of 12 Days. Aerospace Med., Vol. 36, No. 8, August 1965, pp. 733-754

16. Guedry, F. E.; Collins, W. F.; and Grabiel, A.: Vestibular Habituation During Repetitive Complex Simulation. A Study of Transfer Effects. Joint Report with NASA (NASA CR-53123: BuMed-93) January 15, 1964

17. Stone, R. W.; and Letko, W.: Some Observations on the Stimulation of the Vestibular System of Man in a Rotating Environment. Symposium on the Role of the Vestibular Organs in the Exploration of Space NASA SP-77, pp. 263-278

18. Guedry, F. E.; Kennedy, R. S.; Herns, C. S.; and Grabiel, A.: Human Performance During Two Weeks in a Room Rotating at Three RPM. BuMed Rept. 74 NASA CR-55622, August 28, 1964

19. Ruch, T. C.: Pontobulbar Control of Posture and Orientation in Space. In: Medicine Physiology (Ruch and Fultin, eds.) W. B. Saunders Co., 1964, pp. 206-219

20. Gillingham, K. K.: Training the Vestibule for Aerospace Operations USAF School of Aerospace Medicine, Brooks AFB, Tex. Review 8-65, September 1965

21. Lansberg, M. P.: The Modulating Influence of the Otolith Organs on Semicircular Canal Functions. Symposium on the Role of Vestibular Organs in the Exploration of Space NASA SP-77, pp. 69-76

22. Sasaki, H.; Yamagata, M.; Watanabe, T.; Ogino, K.; and Othara, S.: On the Function of the Otolithorgans. In: Contributions to the Vestibular Physiology and Vestibular Test. Acta Otolaryng. Suppl. 179, Tokyo 1963, pp. 42-55

23. Grabiel, A.: Oculogravic Illusions Arch. Opthal. (Chicago) 48:605, 1952

24. Clark, B.; and Grabiel, A.: Factors Contributing to the Delay in the Perception of the Oculogravic Illusion. Joint Report with NASA, NSAM-944, August 1965

25. Walsh, E. G., The Perception of Rhythmically Repeated Linear Motion on the Horizontal Plane, Brit. J. Psychol., 53:1962, pp. 439-445

26. Walsh, E. G., The Role of the Vestibular Apparatus in the Perception of Motion on a Parallel Swing, J. Physcol., 155, 161, pp. 506-513

27. Bergstedt, M., Vestibular Clinical Examination With Use of Electronystagmography, Aerospace Med., September 1964, pp. 884-887

28. Johnson, W.H., The Importance of the Otoliths in Disorientation, Aerospace Med., September 1964, pp. 874-877

29. Miller, E.F., Counterrolling of the Human Eyes Produced by Head Tilt With Respect to Gravity, Acta otolaryng. Stockholm 54, 1961, pp. 479-501

30. Miller, E.F.; Grabiel, A.; and Kellogg, R.S.: Otolith Organ Activity Within Earth-Standard One-Half Standard and Zero Gravity Environments, Joint Report with NASA, NSAM-943, August 1965

31. Guedry, F.E., and Harris, C.S., Labyrinthine Function Related Experiments on the Parallel Swing, Joint Report with NASA (NASA Order No. R-93) (BuMed-86) September 1963

32. Grabiel, A., and Fregly, A.R., A New Quantitative Ataxia Test Battery, Symposium etc., pp. 99-119

LIBRARY CARD ABSTRACT

The need for a device to determine the mass of man and other animate and inanimate objects in the zero gravity environment of space is of importance in ascertaining the physical condition of the flight crew and the successful completion of certain scientific experiments to be conducted in future space vehicles. This report defines the study, design, fabrication and test of such a device.